DESIGN AND COMPARATIVE PERFORMANCE OF 1-1 COUNTER AND SPLIT-FLOW HEAT EXCHANGERS

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CERTIFICATE

This is to certify that the work entitled "Design and Comparative Performance of 1-1 Counter- and Split-flow Heat Exchangers" by Lok Ranjan is a record of work carried out under my supervision and has not been submitted elsewhere for a degree.

June, 1987

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Lok Ranjan

CONTENTS

LIST OF FIGURES

NOMENCLATURE

A	-	Total heat transfer area (m^2)
a_t	-	Flow area for a single tube (m^2)
b	-	Baffle spacing (mm)
c	-	Specific heat for tube-side (cold) fluid (J/kg.C)
C	-	Specific heat for shell-side (hot) fluid (J/kg.C)
c_b	-	Clearance between the tube in baffle (mm)
d	-	Diameter of the tube (mm)
D	-	Diameter of the shell (mm)
f	-	Friction factor
G	-	Mass velocity ($kg/m^2.s$)
h	-	Film heat transfer coefficient ($W/m^2.C$)
h_{io}	-	Inside film heat transfer coefficient referred to outside area ($W/m^2.C$)
j_H	-	Heat-transfer factor
k	-	Conductivity of fluids (W/m.C)
L	-	Length of tubes (m)
m	-	Tube-side mass flow rate (kg/s)
M	-	Shell-side mass flow rate (kg/s)
n	-	Number of crosses on the shell-side
N	-	Number of tubes
n_b	-	Number of baffles
p_T	-	Pitch of the tubes in a layout (mm)
ΔP	-	Pressure drop (mm of water or N/m^2)
Q	-	Overall heat transfer (kW)
R	-	Ratio of heat capacity of cold fluid to hot fluid

R_d	-	Dirt factor for fouling (m^2.C/W)
t	-	Cold fluid temperature (C)
Δt	-	True temperature difference (C)
T	-	Hot fluid temperature (C)
T'	-	Hot fluid temperature at end of counterflow section (C)
T"	-	Hot fluid temperature at end of parallel flow section (C
U	-	Overall heat transfer coefficient (W/m^2.C)
U'	-	Modified overall heat transfer coefficient (W/m^2.C)
μ	-	Viscosity for the fluids (N/m^2.s)

Subscripts

a	-	Refers to air
b	-	Refers to values for the baffle
cf	-	Refers to counterflow section
c	-	Refers to clean value
d	-	Refers to design values
e	-	Refers to equivalent values
et	-	Refers to ethyl alcohol
i	-	Refers to inside values
o	-	Refers to outside values
m	-	Refers to modified values
p	-	Refers to parallel flow section
s	-	Refers to shell-side
sf	-	Refers to split-flow values
t	-	Refers to tube-side
v	-	Refers to water
w	-	Refers to wall
x	-	Refers to quantities at a location x

1	-	Refers to inlet side
2	-	Refers to outlet side

ABSTRACT

A computer program has been developed for the design of 1-1 counter- and split-flow heat exchangers. The effect of dividing the flow upon the performance of the counterflow arrangement has been studied theoretically and experimentally.

The shell-side theoretical pressure drop in split-flow has been found out to be 0.1309 to 0.1328 times that in counterflow. However, the experimental value for this ratio is in the range of 0.25-0.31.

The theoretical predictions of the reduction in heat duty are 5-12% in case of split-flow as compared to counterflow arrangement for the same inlet conditions. This is in quite close agreement with the experimental value of 10-15%.

The computer program also designs and evaluates alternative split-flow configurations, where this reduction is compensated by increasing the exchanger area; either by increasing the length of the tubes or their number. The superiority of either of these alternative designs is dependent upon the desired specifications.

CHAPTER - 1

INTRODUCTION

1.1 Description

Heat exchangers are devices employed commonly to transfer thermal energy from one fluid to another. The wide range of heat transfer services required in the energy, process, environmental control and transportation industries has led to the development of an equally wide variety of heat exchanger configurations. The shell-and-tube type exchangers are extensively used because of their rugged construction, the great range of design variations and operating conditions that can be accommodated, fairly high performance and other advantages of practical significance [1, 2].

The design of heat exchangers may involve alterations specific to the applications with the broad outline being often the same. For applications falling in the general category, the design procedure can be quickly adapted to a form suitable for digital computation. The manual design involves repeated references to charts and tables for data collection which can be handled accurately and efficiently by a digital computer.

The counter-flow configuration which gives the maximum heat exchange per unit length is preferred but it is not always feasible. There are instances where the fluid flow rather than the heat transfer is the controlling factor. It may not be possible to meet the allowable pressure drop by conventional methods. The examples of it are found when:

(i) either the true temperature difference or overall heat transfer coefficient is very large.

(ii) one fluid stream has a very small temperature range as compared to the other.

(iii) the allowable pressure drop is small [3].

In such cases, the split-flow configuration is used where the reduced mass velocity and shell-side path length lead to a substantial reduction in the shell-side pressure drop. The magnitude of pressure drop in a split-flow configuration has been taken to be $\frac{1}{8}$ of that in a counter-flow situation. But this approximation neglects the change in the friction factor with Reynolds number, which could be quite significant. In addition, the heat duty is also reduced in a split-flow configuration, due to reduction in overall heat transfer coefficient and the mean temperature difference as well. If efforts are made to compensate for the reduction in heat duty, the magnitude of the pressure drop on the shell-side comes out to be even higher than just $\frac{1}{8}$ of the counter-flow configuration.

The economic aspects of such an effect are reflected in the higher than anticipated running costs alongwith the increased initial and maintenance costs. Other factors, as the increased material and space requirements should also be weighed in comparison with the actual advantage accruing, due to a reduced magnitude of pressure drop derived by the split-flow exchanger and only then should they be recommended.

1.2 Review

The earliest systematic efforts to study heat transfer characteristics in a shell-and-tube configuration can be credited to Colburn [4,10]. Its validity was restricted to Re > 2000 and staggered lay-outs. Later on a factor of 0.6 was suggested for baffled shells, for bypass and leakages.

The complexity of the shell-side flow makes the analysis of flow pattern difficult. Attempts were made by Tinker [5,10] to analyse the flow pattern by taking into consideration not only the effective cross flow streams but also the leakage streams in:

(i) the orifice formed by the clearance left between baffle tube-hole and the tube-wall.

(ii) the gap between the bundle and the shell-wall.

(iii) the gap between baffle edge and the shell-wall.

(iv) the tube pass partitions if present in the lay-out.

The model so developed is quite complex and does not lend itself to easy understanding and implementation in actual design procedures.

A largely simplified approach of the Tinker's method is followed in the Bell-Delaware method [6,10]. This is a semi-analytical, non-iterative procedure. It simplifies the Tinker's flow model by assuming ideal tube bank flow and neglecting the interaction between the other streams. In spite of the simplicity and reasonable accuracy, it needs a large and comparatively complex set of data. In addition, certain values are to be estimated from experience for the procedure to begin, which is not very desirable.

The stream analysis method [7,10] applies Tinker's flow model rigorously to the design procedure and is extremely accurate. But the data input required is extensive and complex and requires fast computers for computation. Also, some of the crucial correlations of flow resistance values are not in public domain.

The numerical prediction methods [8,9,10] predict the shell-side flow pattern by solving the flow equation numerically, for a mesh suitably selected to describe the shell. Once the velocity is specified, the heat transfer coefficient may also be calculated on a local basis. This method is recent and promising, but difficult to apply for complex cases. For design purpose, it is not yet a substitute for the other established methods.

The integral apporach as suggested by Kern [3] has been used as a virtual standard in industry. Though the method does not account for leakage and by-pass, the overall design problem is approached as an entity. The predictions of pressure drop and heat transfer rate are reasonably accurate and on the safer side. This method gives maximum insight into the procedure of design.

The split flow heat exchangers are preferred when the allowable pressure drop cannot be met by a counterflow configuration. The shell-side pressure-drop for the 1-1 split-flow exchanger is found to be 0.125 times that of a counterflow arrangement [3,11]. But the analysis does not include the effect of the change of Reynolds number upon the friction factor and the heat-transfer coefficient on the shell-side.

Recently an attempt has been made to investigate these effects analytically for various flow regimes [12].

1.3 Present Work

It involves the design of a 1-1 counter- and split-flow exchangers for a specified application. This enables a direct comparison between the performances of the two configurations in terms of heat exchange and the pressure drop.

The computational procedure follows a modified Kern's approach, the main modifications being:

(i) The charts and tables have been replaced by equations and correlations, some from standard texts [13,14] and the others obtained by a suitable curve fitting.

(ii) The variations with the temperature in the properties of the fluids are accounted for.

(iii) The true temperature difference for the split flow configuration has been evaluated separately rather than considering it to be the same as that for counterflow.

(iv) The computational procedure of the program is more complex (e.g. use of 'nested' iterations) for more accurate results.

A more realistic analysis of the effect of dividing the flow has been considered and the ratio of the shell-side pressure-drop in split flow to that in counterflow has been found to be 0.147.

Also, the decrease in the shell-side heat-transfer-coefficient implies a subsequent reduction in the overall heat transfer-coefficient causing a fall in the heat-duty of the exchanger. This effect has also been considered while arriving at the design.

Both of these effects have been experimentally investigated. The set-up is designed and fabricated for the purpose,

having provisions for counter- as well as split-flow measurements of shell-side pressure-drops and temperature variation in the two configurations. The experimental value for the ratio of the pressure drops on the shell-side is found to be 0.25 to 0.30.

CHAPTER - 2

HEAT EXCHANGERS

Heat exchange between moving fluids is accomplished in an apparatus or equipment called heat exchanger. It has diverse applications in industries leading to innumerable configurations and types of designs. Some important bases of classification are as follows [15] :

(i) the flow configuration,

(ii) the mode of interaction between the streams passing through the heat exchanger,

(iii) the temperature change pattern,

(iv) the interface between the streams,

(v) the function of the equipment, and

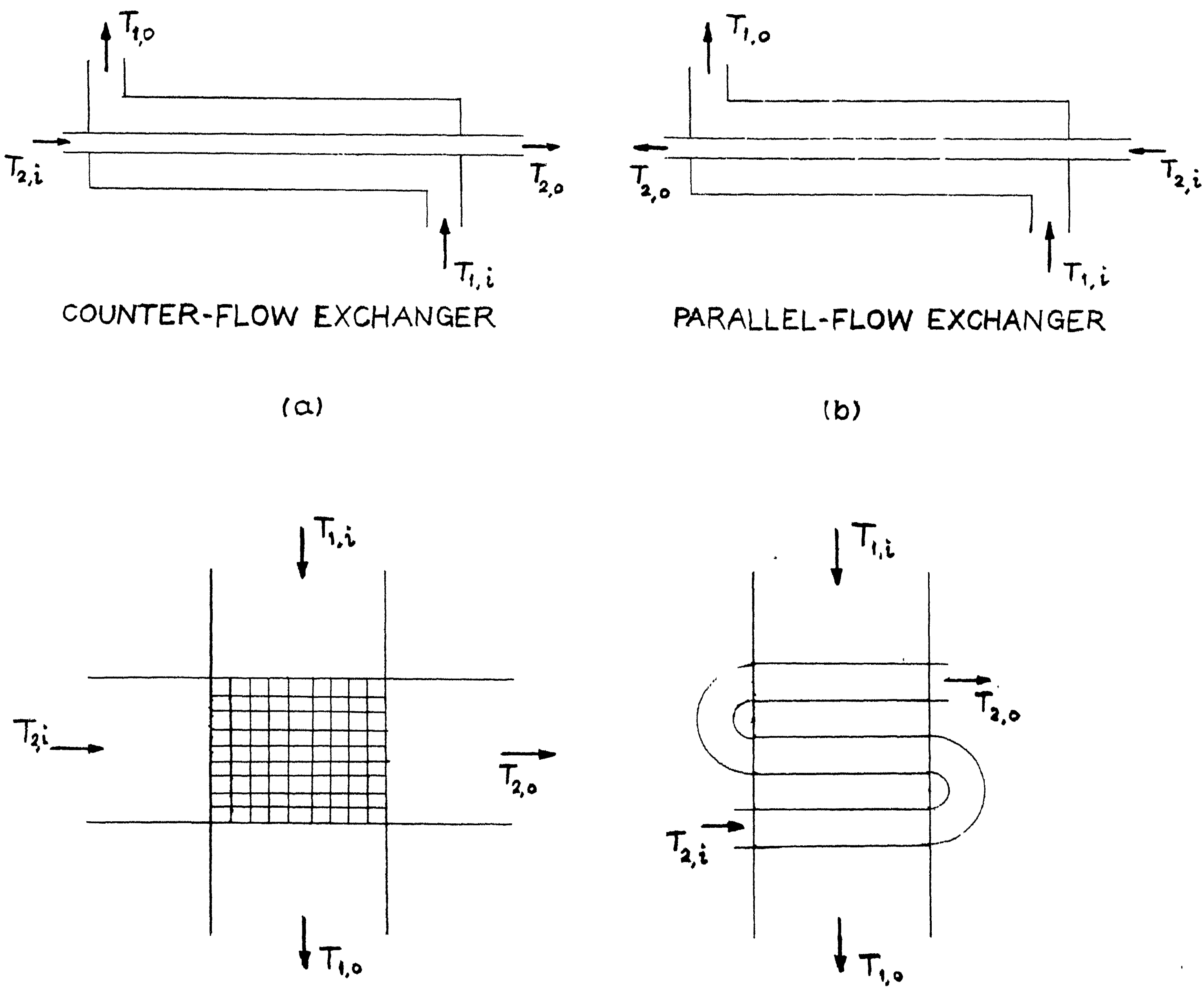
(vi) the mode of time dependent variation.

These are described briefly in the following sections of this chapter.

2.1 Types of Flow Configurations

A major characteristic of the heat exchanger design is the relative flow configuration. This deals with a set of geometric relationships between the streams. These are the ideal representations and the actual flow pattern can never be made to conform exactly to these idealizations. The various possible configurations are:

2.1.1 Counterflow: The two streams of fluid flow parallel to each other but in opposite directions, Figure 2.1(a). Such



COUNTER-FLOW EXCHANGER

(a)

PARALLEL-FLOW EXCHANGER

(b)

CROSS-FLOW EXCHANGER

(c)

CROSS-COUNTER-FLOW EXCHANGER

(d)

SCHEMATIC REPRESENTATIONS OF DIFFERENT HEAT-EXCHANGERS

FIG. 2.1

exchangers are the most efficient, making the best use of the available temperature difference and can render the maximum change in temperature of each fluid stream.

2.1.2 Parallel flow: Figure 2.1(b) exhibits the two fluids, flowing parallel to each other in the same direction. Their utilization of available temperature difference is relatively poor and hence they are not employed when efficiency is the most important factor in design. But they have more uniform wall temperatures. Such configuration is used in superheaters of thermal power plants.

2.1.3 Cross-flow: The two streams flow at right angles to each other, Figure 2.1(c). They are intermediate, in efficiency, between parallel flow and counter-flow exchangers. It is often easier to construct than either of the two, due to practical reasons of ducting of the fluids towards the heat transfer surface.

2.1.4 Cross-counter-flow: The configuration shown in Figure 2.1(d) is termed as cross-counter-flow exchanger. The number of passes depends upon particular requirement. This arrangement compromises between the desired aspects of efficiency and ease of construction. The greater the number of passes, the closer is the approach to counter-flow economy.

2.1.5 Multipass shell and tube: The features of parallel and counter-flows are combined by either bending the tubes or using straight tubes with suitably sub-divided headers. The hairpin arrangement is easier to construct as only one end of the shell needs to be perforated.

There may be configurations in which the shells are coupled together, Figure 2.2(a).

2.1.6 Split flow: The shell-side inlet nozzle is located at the centre of the shell rather than the end of the shell and two outlet nozzles are used, Figure 2.2(c). The number of passes can be multiple. The flow relation is neither true counter-flow nor true parallel flow, and the temperature utilization is less efficient than the counter-flow. But the shell-side pressure drop is extremely low as compared to other configurations.

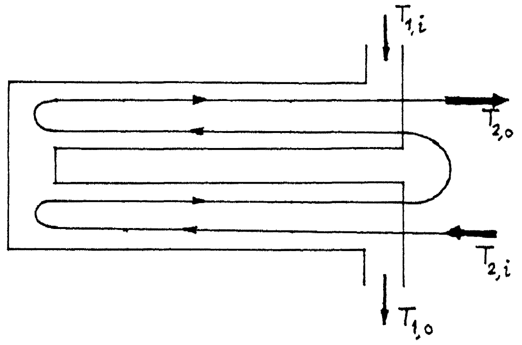
2.1.7 The general case: Figure 2.2(b) shows a multiple flow in inter-penetrating continua, of which the above flows are particular examples. There may be several entry and exit points for different streams which may sub-divide after entry and re-combine before exit. The heat exchange occurs when the individual streams come into thermal contact within the exchanger volume.

2.1.8 Regenerators: In all the above cases, the flows are steady and simultaneous with different streams flowing in different spaces. When the streams flow alternately through the same space, the exchanger is known as regenerator, Figure 2.2(d). Here heat transferred by one fluid is stored in the walls of the duct and is given to the second fluid when it, in turn, flows past.

These can assume any one of the above configurations. They are periodic flow devices and often involve rotary valves at each end to control the flow alternately.

2.2 Types of Interaction between Fluid Streams

The interactions between the fluid streams in an exchanger

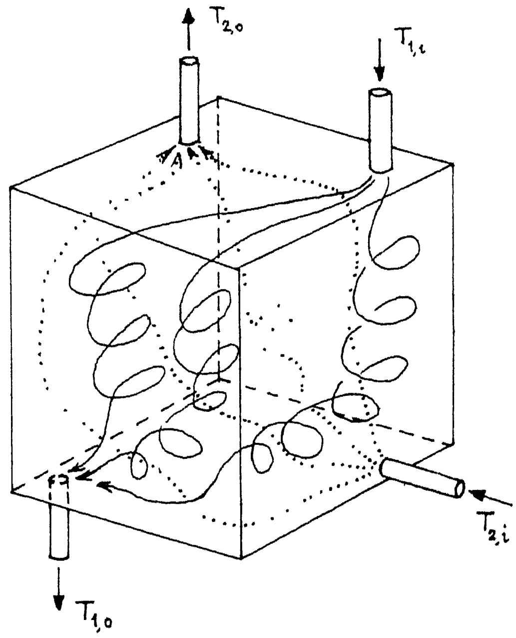


A 2-SHELL PASS, 4-TUBE PASS TYPE HEAT EXCHANGER

A typical example of :

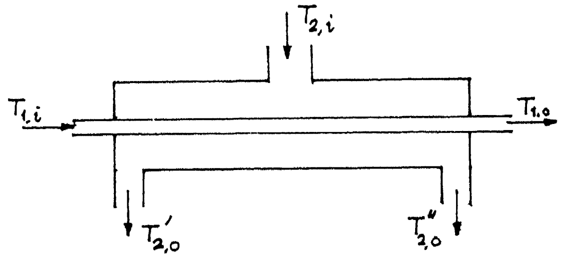
MULTI-PASS EXCHANGER

(a)



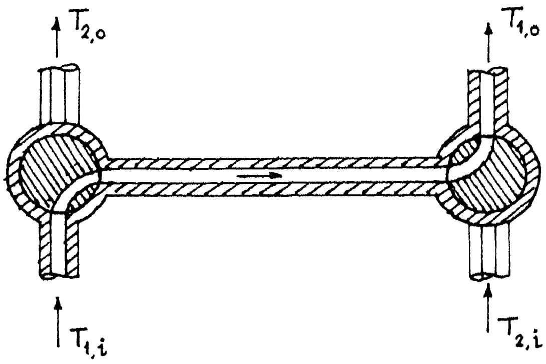
GENERAL CASE

(b)



SPLIT-FLOW EXCHANGER

(c)



REGENERATIVE EXCHANGER

(d)

SCHEMATIC REPRESENTATIONS OF DIFFERENT HEAT-EXCHANGERS

FIG. 2.2

may involve either exclusive heat transfer or a combined simultaneous heat and mass transfer. The heat transfer may, in its turn, be simultaneous or non-simultaneous. Often, the heat exchange is more important consideration though the significance of mass transfer cannot be overlooked.

2.2.1 Heat transfer: It occurs because of temperature difference between streams. It may be effected indirectly between the fluids separated by a dividing wall. The heat transfer may entail phase change of any of the fluids. Such exchangers are termed as recuperators when the flow of fluids is simultaneous and regenerators when the flow of fluids is alternate.

Heat transfer can also occur when the fluids are in direct contact. For non-phase change cases, the mass transfer is absent. Cooling towers are such devices. Sometimes, as in fluidized bed heat exchanger, a cloud of solid particles exchanges heat with a stream of fluid. The solid particles are in a semi-random motion but are prevented by gravity from rising with the gas. These may exchange heat with solid surfaces carrying fluid as well.

2.2.2 Simultaneous heat and mass transfer: This is effected by way of vaporization of a part of liquid stream or the condensation of a component of the gaseous stream, resulting in heat and mass transfer. The phase change is accompanied by thermal effects due to latent heat of phase change apart from the sensible heat exchange.

The effectiveness of heat exchange may be increased manifold in these cases, for example, the wet cooling towers involving

involving mass exchange may require less than 20% of area of contact as compared to dry cooling towers. These interactions are measured in terms of volumetric or superficial interaction coefficients viz. heat and mass transfer coefficients.

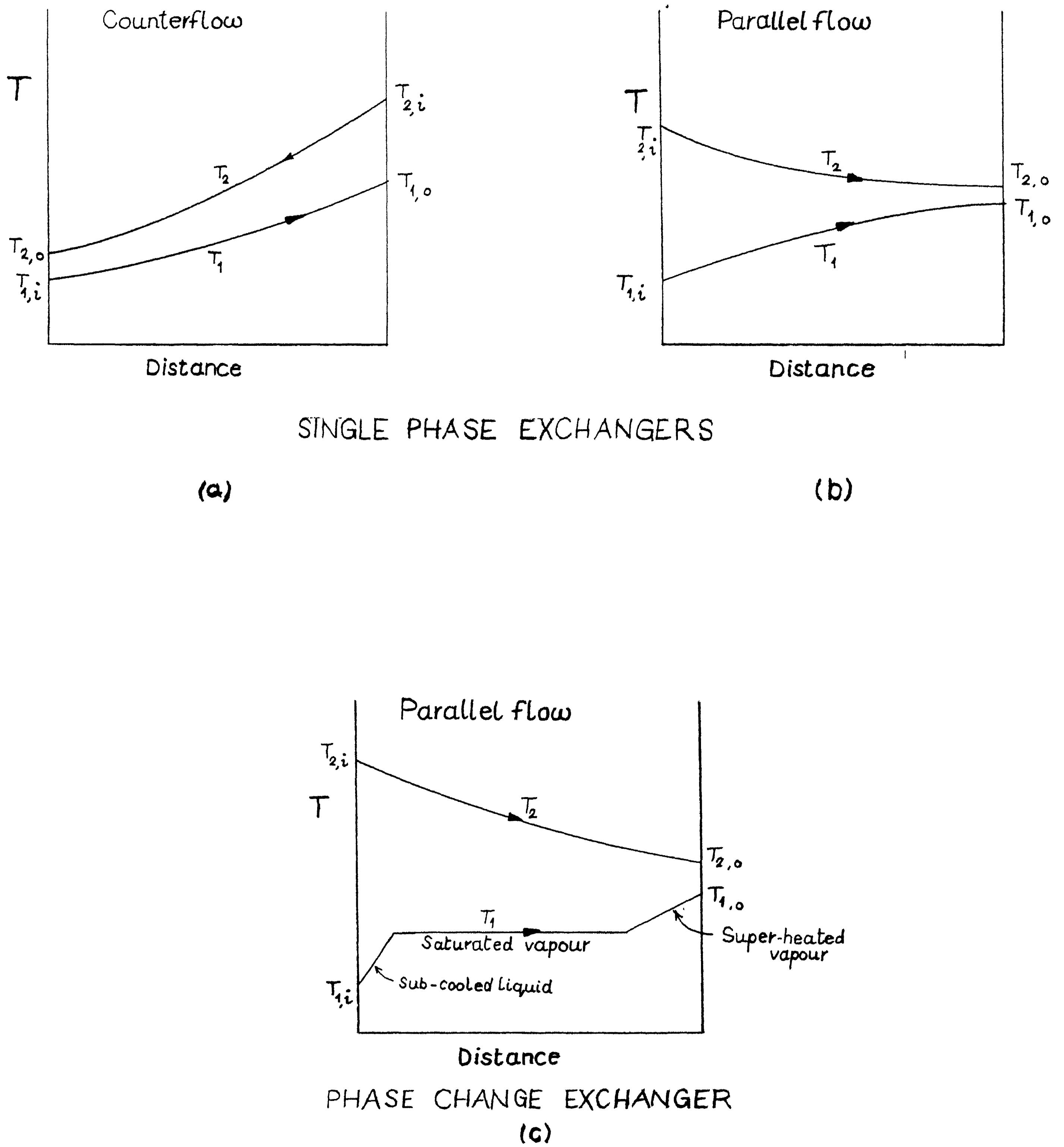
2.3 Temperature Change Profiles

In general, the temperature of a fluid stream varies in a non-linear fashion. The non-linearity may sometimes be insignificant, yet it is present always. The non-linearities are results of change in properties and make the analytical approach difficult, therefore necessitating numerical techniques.

The pattern of temperature variation may in addition be continuous or discontinuous depending upon the absence or presence respectively of phase change.

2.3.1 Non-phase change exchangers: Each fluid stream leaves the exchanger in the same phase in which it enters. The temperature of the cold fluid increases while that of hot fluid decreases in a continuous manner (provided the variation in properties is continuous) with the distance, Figures 2.3(a) and 2.3(b).

2.3.2 Phase change exchangers: In a general case, a sub-cooled liquid may enter as a cold stream, undergo sensible heat addition, then undergo a change of phase with no temperature variation and then be superheated. The inverse process for a superheated gas is also possible. The variation of temperature in these cases is discontinuous, Figure 2.3(c).



TEMPERATURE-CHANGE PATTERNS FOR DIFFERENT EXCHANGERS

FIG. 2.3

2.4 Interfaces between Streams

The fluid streams in a heat exchanger are brought into direct or indirect contact in a wide variety of ways. This leads to several different fluid interfaces. Some common ones are as follows:

2.4.1 Plain tubes: One of the fluids flows in a straight or curved tube, mostly of a circular cross-section whereas the second fluid flows outside these tubes in a longitudinal, perpendicular or oblique direction. The interface in such cases is represented by the tube wall and the contact is indirect.

2.4.2 Finned tubes: When the heat exchange is effected much more easily on one side of the tube, the other surface is extended by provision of fins. Mostly the fins are provided on the outside of the tubes; occasionally it is the inside of the tube, when the internal heat transfer coefficient is lower.

The fins may be integral with the walls or may be attached to it. The presence of fins entails that, per unit volume of exchanger, there is more interface area between metal and fluid for one stream than the other.

2.4.3 Matrix arrangements: These are configurations devised to increase the interfacial area which, in turn, increases the interaction coefficients. These prove economical ways of increasing heat transfer and are termed as heat exchange matrices. Their structure is significantly different from the tube arrangements to increase area.

2.4.4 Films: There are several instances in which direct contact

exists between a stream of gas and a film of liquid, that runs on a solid surface. The film forms the interface in such cases, the examples being gas-absorption towers and cooling towers in power stations.

2.4.5 Sprays: In these cases, fine droplets of liquid interact with a stream of gas to effect heat and mass exchange. The droplet surface forms the interface and such cases often involve mass exchange as well. Common examples are desuperheaters and humidifiers.

2.5 Types of Heat Exchange Equipment

The heat exchange equipment can be classified on the basis of either the function that it performs or its construction.

2.5.1 Functional classification: A heat exchange equipment can be categorised into any one of the following class of equipments on the basis of its application [16,17].

Chiller : cools a fluid to a temperature below that obtainable if water only were used as a coolant.

Condenser : condenses a vapour or a mixture of vapours, either alone or in the presence of non-condensing gases.

Partial condenser: condenses vapours at a point high enough, to provide a temperature difference sufficient to preheat a cold stream of process fluid. This eliminates the need for providing a pre-heater.

Final condenser: condenses the vapour to a final storage temperature.

Cooler : cools liquids or gases by means of water.

Cooling tower : cools the water for recirculation by contact with air. The contact may be direct or indirect.

Dryer : vaporizes liquid from a mixture of highly wetted solid and the liquid.

Exchanger : performs double function of cooling a hot fluid and heating a cold one without loss of heat.

Fired heater : uses a hot stream generated by combustion within the exchanger.

Heater : imparts sensible heat to a liquid or gas.

Reboiler : provides the heat necessary for distillation in a fractionating tower by using steam or a hot process fluid.

Steam generator: generates steam for use elsewhere by using high level of heat.

Superheater : heats a vapour above the saturation temperature.

Vaporizer : causes a part of liquid to evaporate by heating.

Waste heat boiler: is a steam generator using a hot gas or liquid produced in a chemical reaction as the heating medium.

This list is by no means exhaustive because of the innumerable uses which a heat exchange equipment can be put to.

2.5.2 Structural classification: The wide range of applications alongwith the specific demands of applications have led to the development of so many exchangers of different constructions, that only a limited listing is possible within the purview of this chapter. This is shown in a chart as follows:

Heat Exchangers

Single phase		Evaporators
For solids	For liquids & gases	
1. Fluidized bed type	1. Shell-and-tube type	1. Forced circulation type
2. Moving bed type	2. Double pipe	2. Short-tube vertical
3. Agitated pan type	3. Plate type	3. Long-tube vertical
	4. Graphite block type	4. Horizontal tube
	5. Spiral tube type	
	6. Air cooled (high-finned)	
	7. Matrix (plate-fin)	

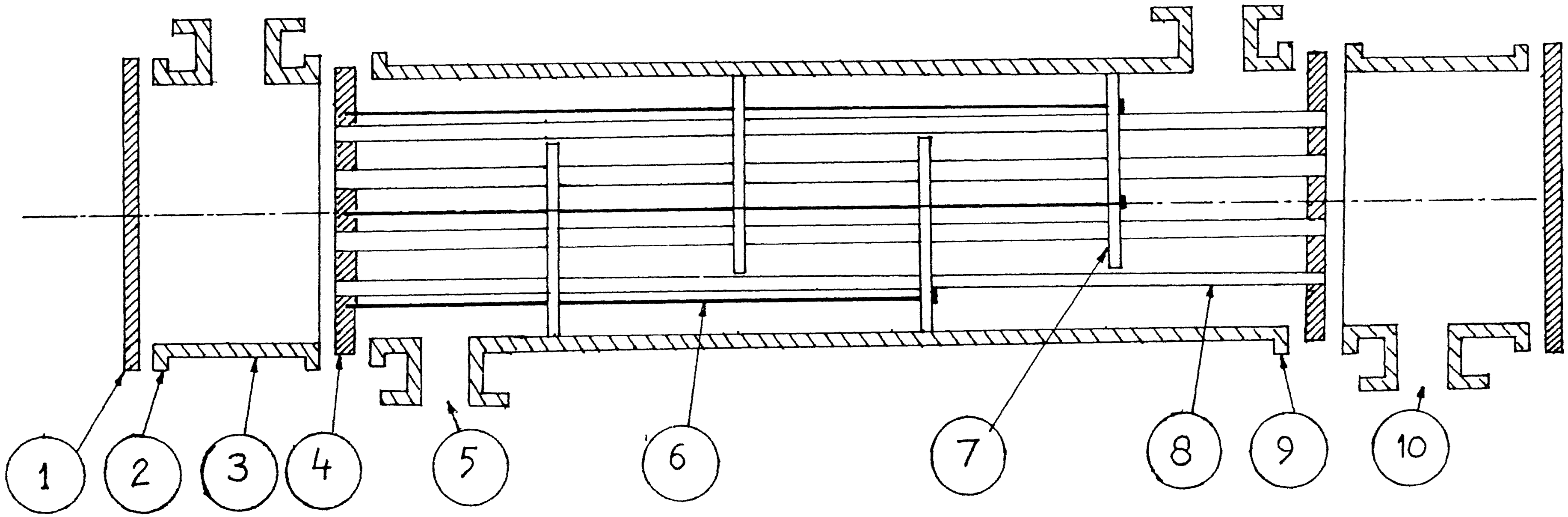
Of all the heat exchangers mentioned, the most versatile and commonly used ones are the shell-and-tube type of exchangers.

2.5.3 Construction of a shell-and-tube heat exchanger: A typical shell-and-tube exchanger is shown in Figure 2.4.

The tubes are the basic components, providing the heat transfer surface between the fluids, one of which flows inside the tube and the other across them. The tubes may be plain or with fins.

These are held in place by a suitable number of tube sheets (one for U-tube design and two for the rest). The tube-sheet is a metal plate that has been drilled and grooved for the tubes and other mating components as gaskets, baffle spacer rods and bolt circle.

The shell is a usually cylindrical structure surrounding the tubes and containing the fluid outside the tubes. It may be made out of pipes or by rolling a metal plate into a cylinder and welding the longitudinal joint.



1. CHANNEL COVER
2. HEAD FLANGE
3. HEAD
4. TUBE SHEET
5. SHELL NOZZLE
6. TIE ROD & SPACER
7. BAFFLE
8. TUBE
9. SHELL FLANGE
10. HEAD NOZZLE

A TYPICAL SHELL & TUBE TYPE HEAT-EXCHANGER

FIG. 2.4

The nozzles are provided on both the shell as well as the tube side for the inlet and outlet ports for the fluid streams. On the tube side, channels are provided for collection and distribution of fluid from the tubes. These are provided with channel covers so that the tubes and the tube-sheet can be exposed for inspection.

The baffle array supports the tubes against bending and vibration and guides the flow back and forth across the tube bundle to increase the heat transfer rate.

Other essential components are tie-rods, spacers, sealing strips, gaskets etc. The details of construction and assembly of the various exchanger components are available in [17].

2.5.4 Reasons for use of shell-and-tube exchangers: The shell-and-tube type of exchangers are versatile and can accommodate various operational and constructional variations. This makes them the most widely employed exchangers. Some advantageous features of this class of exchangers are as follows [2]:

(i) they have fairly large ratio of heat transfer area to volume and weight.

(ii) they are easy to construct, rugged and can use a great variety of materials as per strength and economy demands.

(iii) they are easy to clean by removal of tubes; also, the components more prove to failure e.g. gaskets and tubes can be easily replaced.

(iv) they are applicable both to single phase and phase change situations with wide temperature range and permissible temperature differentials. The pressure range is from

vacuum to high values and the range for permissible pressure drops is also very vast.

(v) they can accommodate thermal stresses rather inexpensively.

(vi) they can easily accomplish the use of enhanced surface, either internal or external.

(vii) they have a wide range of size.

(viii) they make possible the positive separation of fluid.

CHAPTER - 3

ANALYSIS AND COMPUTATIONAL PROCEDURE

3.1 Introduction

As discussed in Chapter 1, the shell-and-tube exchangers are very versatile and are available with many variations in configuration and design. An elaborate discussion or design of even a fraction of these is not possible in a brief investigation as this. The emphasis in this study is on the effect of velocity and the Reynolds number of the flow upon the heat transfer and pressure drop characteristics; especially on the relative actual performance of a split-flow arrangement vis-a-vis a counterflow arrangement.

For this purpose the study of a simple 1-shell pass, 1-tube pass arrangement has been undertaken in the place of other more complex arrangements. In the latter, the effects are no doubt present but are screened somewhat due to the presence of various complicating factors, especially in case of heat transfer for split-flow arrangement.

3.2 The Approach to Design

The logical process followed in the design of heat exchanger has been outlined in Figure 3.1. The dotted lines represent the steps to assess the relative performance of a split-flow configuration.

The method used to perform the desired steps within the domain of program is the shell-side method based on integral

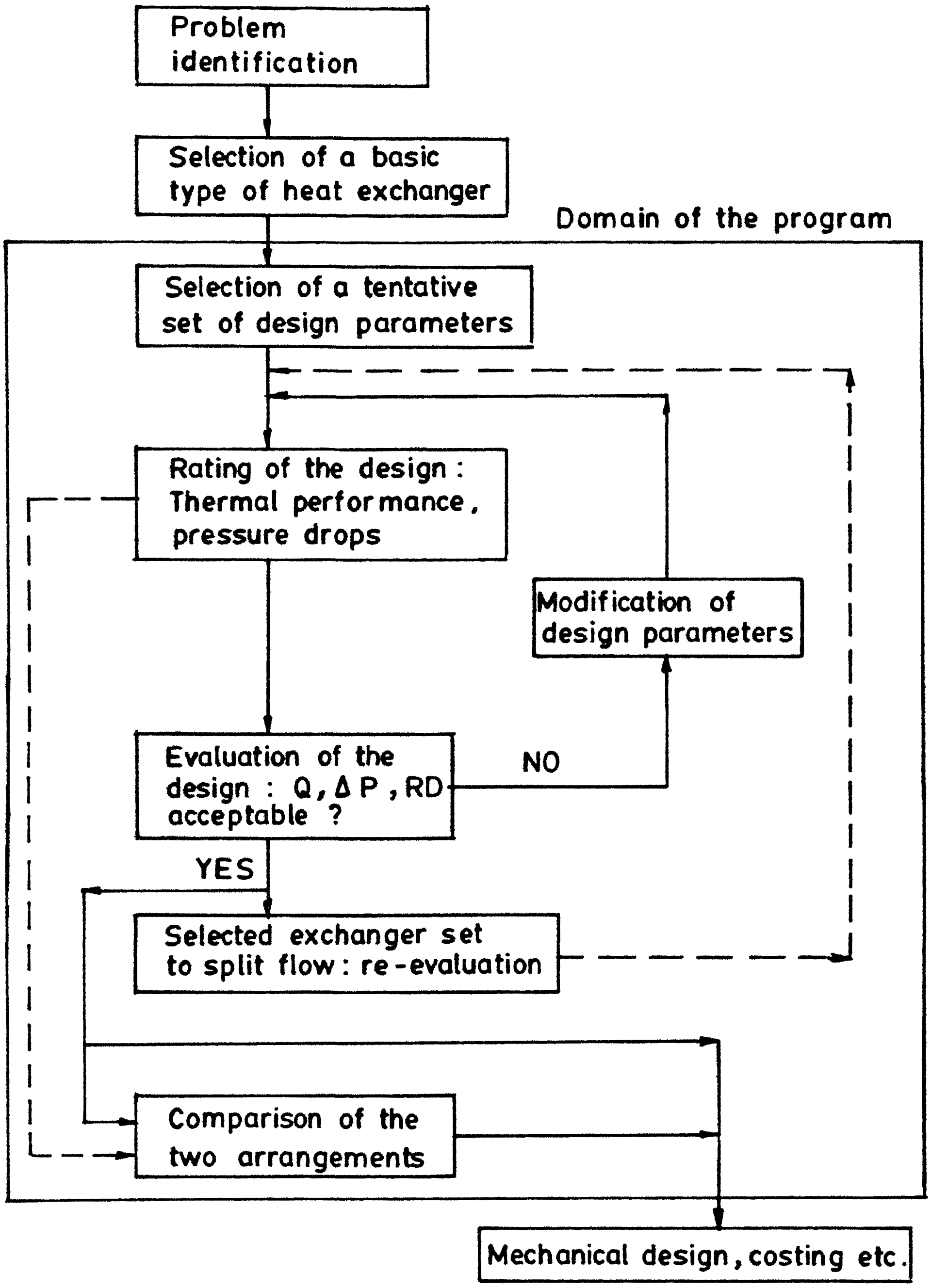


FIG. 3.1 OUTLINE OF THE PROGRAM

approach as suggested by Kern [3]. The equations used in this approach are based upon the overall data for actual baffled exchangers. The main argument in favour of the suitability of this approach, rather than the more recent analytical methods, is that it provides a direct comparison of the heat-transfer performance and the shell-side pressure drop in the 1-1 counter- and split-flow arrangements. This aspect has gone largely unnoticed so far in literature.

In addition, the method is simpler requiring lesser number of data as input. The data needed is fully definite rather than being based on judgement as certain data for analytical methods are. Moreover the method is safe from pressure drop point of view. This might in some cases lead to a lower heat transfer performance, but the effect of this drawback is sought to be minimized by using as many effective passes as permissible.

3.3 Mathematical Modelling

The mathematical modelling for the heat exchanger is elaborated below stepwise as depicted in Figure 3.1.

3.3.1 Selection of basic type of exchanger to be designed: The exchangers selected in the present study are 1-1 counter- and split-flow arrangements.

3.3.2 Selection of a tentative set of exchanger design parameters: Heat balance equation is used to evaluate the terminal temperatures and the true temperature difference for counterflow. The caloric temperature for the fluids under consideration (air or water) in this study can be taken to be the same as the mean temperature (Appendix 3).

The initial design value of the overall heat transfer coefficient U is then assumed, depending upon the application. For the known temperature difference and assumed U, the area is evaluated and the nearest tube count is obtained. Thereafter the design value of U is slightly modified.

$$Q = MC(T_1 - T_2) = mc(t_2 - t_1) \tag{3.1}$$

and

$$\Delta t = \frac{(\Delta t_2 - \Delta t_1)}{\ln(\frac{\Delta t_2}{\Delta t_1})} \tag{3.2}$$

where $\Delta t_2 = T_1 - t_2$, $\Delta t_1 = T_2 - t_1$

If U is assumed as U_d then

$$A_d = \frac{Q}{U_d \, \Delta t} \tag{3.3}$$

Depending upon the specification, the number of tubes can be obtained as

$$N_d = \frac{A_d}{A_o} \tag{3.4}$$

The actual number of tubes selected is N_t which is the nearest to N_d for the layout. Then U_d is modified to

$$U_m = U_d \cdot \frac{N_d}{N_t} \tag{3.5}$$

The baffle spacing is selected such as to ensure the maximum odd number of crosses in half-shell.

3.3.3 The rating of the design: The tentative design thus obtained is evaluated on the basis of its thermal performance and pressure drop.

The overall flow area on the tube and shell-side is evaluated as

$$A_t = N_t a_t$$

and

$$A_s = (D * c_b * b)/p_T \tag{3.6}$$

The mass velocities are obtained on the two sides as

$$G_t = m/A_t$$

$$G_s = M/A_s \tag{3.7}$$

The Reynolds number for the flows are obtained from

$$Re_t = \frac{d_t G_t}{\mu}$$

$$Re_s = \frac{D_e G_s}{\mu} \tag{3.8}$$

where the property values μ are evaluated at mean temperature and D_e is the equivalent hydraulic mean diameter, Figure 3.2(a), defined as:

$$D_e = \frac{(4 \text{ x free area})}{\text{wetted perimeter}}$$

$$= \frac{4 \text{ x } (p_T^2 - \pi d_o^2/4)}{\pi d_o} \quad \text{for square layout}$$

and

$$= \frac{4 \text{ x } (\frac{1}{2} p_T \text{ x } 0.86 p_T - \frac{1}{2} \pi d_o^2/4)}{\frac{1}{2} \pi d_o} \quad \text{for triangular layout} \tag{3.9}$$

The heat transfer factor j_H on each side is calculated from the

correlations (see Appendix 6). Since the variation in properties of air and water is small with temperature, the relevant properties can be evaluated at the mean temperature and the film coefficients are obtained from

$$h_i = j_{Hi} \frac{k}{d_t} \left(\frac{c\mu}{k}\right)^{1/3}$$

$$h_o = j_{Ho} \frac{k}{D_e} \left(\frac{C\mu}{k}\right)^{1/3} \qquad (3.10)$$

The inside coefficient is corrected to refer it to the outside area

$$h_{io} = h_i \frac{d_t}{d_o} \qquad (3.11)$$

The clean overall heat-transfer coefficient is then found from

$$U_c = \frac{h_{io}\ h_i}{(h_{io} + h_i)} \qquad (3.12)$$

For evaluating the pressure drop on each side:

(i) the friction factor f for each side is calculated

(ii) the equivalent length L_e on the tube-side is the tube-length L, whereas on the shell-side it is the product of the number of crosses n and the shell diameter D

(iii) the density S on each side at mean temperature is found.

The pressure drop is then evaluated as

$$\Delta P = 4.f \frac{L_e}{D_e} \left(\frac{1}{2} \frac{G^2}{S}\right) \qquad (3.13)$$

where L_e = L on tube-side

$= (n_b + 1)D$ in counterflow on shell-side

$= \dfrac{(n_b + 1)}{2}$ D in split-flow on shell-side.

The dirt factor is calculated from

$$R_d = \frac{(U_c - U_m)}{U_c\ U_m} \tag{3.14}$$

3.3.4 Evaluation of the design: The design thus obtained is checked for satisfactory performance on the following counts:

(i) The heat duty requirement should be met.

(ii) There should be some allowance for the resistances due to fouling.

(iii) The pressure drop requirements should be satisfied.

For satisfying (i) the number of tubes is selected so as to provide sufficient surface area for heat exchange.

To account for fouling, the value of dirt factor should be positive and slightly greater than the desired value. If the value of R_d is negative, a lower value of U is assumed and the design procedure repeated until a satisfactory value (a typical value for $\frac{U_m}{U_c}$ being 0.65-0.7) is obtained. On the other hand, if the value of R_d is very large the starting value of U is increased and the procedure of Section 3.3.3 is repeated until R_d is of the desired order.

Another unsatisfactory arrangement could result when either the tube-side or the shell-side pressure drop exceeds the allowable value. In the case of tube-side, the number of tubes is increased (though it reduces the outside pressure drop and heat transfer as well) whereas for the shell-side,to reduce the pressure drop, the baffle spacing is increased slowly in order to bring down the pressure drop and yet maintain the maximum possible heat transfer conditions.

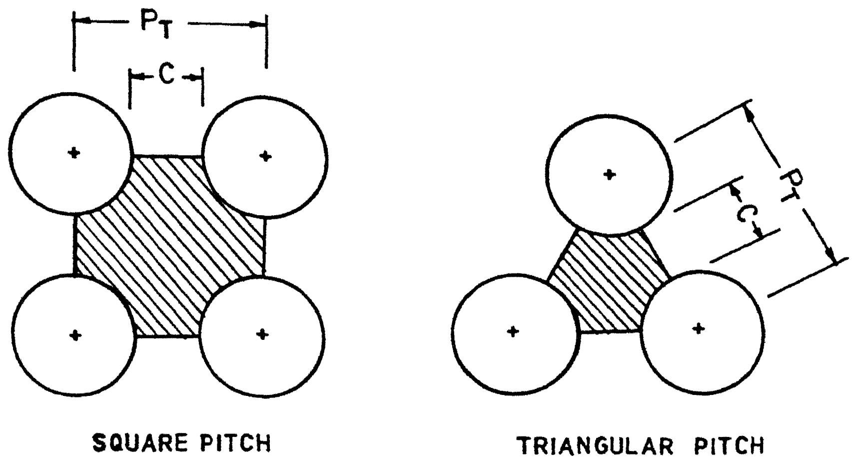


FIG. 3.2 (a) EQUIVALENT DIAMETER

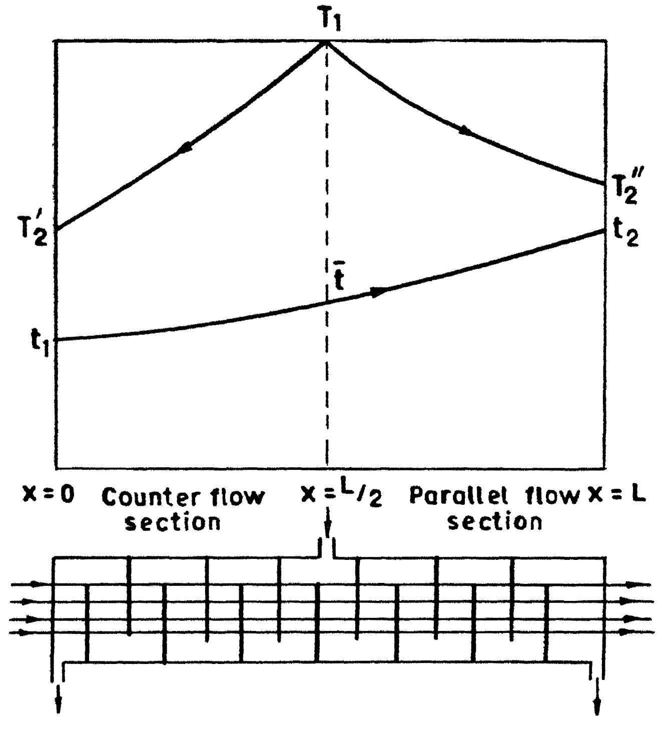


FIG. 3-2 (b) SPLIT FLOW ARRANGEMENT

3.3.5 Split flow heat exchangers: Figure 3.2(b) shows a typical temperature distribution for the split-flow exchanger. The performance of this exchanger is to be compared, to the exchanger obtained after satisfying the requirements for the counterflow, subject to the following constraints:

(i) the inlet temperatures of both fluids are the same as that in the counterflow.

(ii) the mass-flow rates of both the fluids are the same.

(iii) the surface area of the tube-bundle is the same.

The terminal temperatures and the temperature distributions in the split-flow case are obtained by an iterative procedure because the set of simultaneous equations is transcendental.

These equations can be arrived at by following the methodology as in [11], after accounting for the different values of U in the counter- and parallel-flow sections. To start with, the value of the exit temperature of the fluid in the tube, t_2, is assumed. The heat balance gives the value of the mean exit temperature T_2 of the shell-side fluid. The individual exit temperatures T_2' and T_2'' are evaluated on the basis of following analysis.

The heat balance equation for any element dx gives:

$$mc\ dt = U\ dA(T - t) \tag{3.15}$$

Considering heat balance from x = 0 to x with the shell-side flow reduced to half in each section, one gets:

$$mc(t - t_1) = \frac{MC}{2}(T - T_2') \tag{3.16}$$

or in differential form:

$$2mc \frac{dt}{dA} = MC \frac{dT}{dA} \tag{3.17}$$

The combination of Eqs. (3.15) and (3.16) and elimination of $\frac{dt}{dA}$ gives

$$\frac{d^2T}{dA^2} - \frac{2UR}{mc}\frac{dT}{dA} + \frac{2U^2R}{(mc)^2}(T - t) = 0 \tag{3.18}$$

For sensible heat exchange, with $R = \frac{mc}{MC}$ Eq. (3.16) gives

$$\frac{(T - T_2')}{(t - t_1)} = \frac{mc}{MC/2} = 2R \tag{3.19}$$

Combining Eqs. (3.17), (3.18) and (3.19) and elimination of variable t, we get

$$\frac{d^3T}{dA^3} - \left(\frac{2UR}{mc}\right)\frac{d^2T}{dA^2} + \left(\frac{U}{mc}\right)^2(2R - 1)\frac{dT}{dA} = 0 \tag{3.20}$$

which has a general solution of the form:

$$T_x = k_1 + k_2\, e^{\frac{UA_x}{mc}(2R-1)} + k_3\, e^{\frac{UA_x}{mc}} \tag{3.21}$$

By substitution of Eqs. (3.19) and (3.21) into Eq. (3.18) and the application of boundary conditions for the counterflow section

$$T = T_2' \quad \text{at} \quad A = 0$$

and $\quad T = T_1 \quad \text{at} \quad A = A/2$

the value of constants k_1, k_2 and k_3 is found out and the solution for the counterflow section is of the form

$$\frac{(T_x - t_x)}{(T_2' - t_1)} = e^{\frac{UA_x}{mc}(2R-1)} \tag{3.22}$$

Similarly, for the parallel-flow section, the solution is:

$$\frac{(T_x - t_x)}{(T_1 - \bar{t})} = e^{-\frac{U'A_x}{mc}(2R+1)} \tag{3.23}$$

The matching of mid-point temperatures from Eqs. (3.22) and (3.23) gives:

$$\frac{\ln \dfrac{(T_1 - \bar{t})}{(T_2' - t_1)}}{\ln \dfrac{(T_1 - \bar{t})}{(T_2'' - t_2)}} = \frac{(2R - 1)\,U}{(2R + 1)\,U'} \tag{3.24}$$

The heat balance for the overall counter- and parallel-flow sections implies:

$$mc(\bar{t} - t_1) = \frac{MC}{2}(T_1 - T_2') \tag{3.25}$$

and $$mc(t_2 - \bar{t}) = \frac{MC}{2}(T_1 - T_2'') \tag{3.26}$$

Equations (3.24) to (3.26) form a set of non-linear simultaneous equations which are solved iteratively. A value of $\bar{t}$ is assumed and T_2', T_2'' solved for, from Eqs. (3.25) and (3.26). It is then seen if the Eq. (3.24) is satisfied. If it is, then the solution is accepted. But if it is not then a new value of $\bar{t}$ is chosen depending upon whether the left hand or the right hand side is greater. This iterative procedure is carried out until a satisfactory solution is obtained.

The true temperature difference is obtained as the sum of individual LMTD as:

$$\frac{(t_2 - t_1)}{(\Delta t)_{sf}} = \frac{(t_2 - \bar{t})(2R + 1)}{(\Delta t)_p} - \frac{(\bar{t} - t_1)(2R - 1)}{(\Delta t)_{cf}} \tag{3.27}$$

Alternatively,

$$(\Delta t)_{sf} = \frac{(t_2 - t_1)}{\ln \frac{(T_2' - t_1)}{(T_2'' - t_2)}} \tag{3.28}$$

All these values are obtained for an assumed value of t_2. Proceeding exactly as the counterflow case (Section 3.3.3), the overall clean heat transfer coefficient is found and the value of the actual heat transfer coefficient is evaluated, assuming the factor R_d to be the same as:

$$U = \frac{U_c}{(1 + R_d U_c)} \tag{3.29}$$

At this stage, it is checked whether the value of t_2 assumed is proper by comparing $UA(\Delta t)_{sf}$ and $mc(t_2 - t_1)$. If the two are not equal, another value of t_2 is chosen. The procedure from Eqs. (3.24) to (3.29) is repeated until two consecutive values of heat duty differ within a preassigned limit. Ultimately, the temperatures and heat transfer coefficient values and the pressure drop on the shell- and tube-sides are obtained and their rating completed.

It was also found that the procedure from Eqs. (3.15) to (3.28) can be replaced by a more elegant and insightful approach as given in Appendix 4.

3.3.6 Comparative performance: For the split flow, the pressure drops and the overall heat transfer are lesser than that for the counterflow configuration. The ratio and percentage reduction in heat duty and pressure drop on the shell-side is evaluated for

the purpose of comparison. The shell-side pressure drop is also estimated when the exchanger is modified to yield the same heat duty. The ways to achieve it are:

(i) increase the length of the tubes in bundle, their number remaining the same.

(ii) increase in the number of tubes, their length remaining the same.

The detailed flow charts and a sample solution are given in Appendix 7 (Figures A.1 to A.9). Figure A.10 shows the temperature distribution for the sample solution.

CHAPTER - 4

EXPERIMENTAL SET-UP AND PROCEDURE

4.1 System Description

In order to study the actual variation in pressure drop across the shell and the heat transfer performance, a testing rig was fabricated and instrumented as shown schematically in Figure 4.1.

Hot air flows on the shell-side and water as the cold fluid on the tube side. It comprises three main sections:

(i) Air flow circuit

(ii) Water flow circuit and

(iii) Heat exchanger.

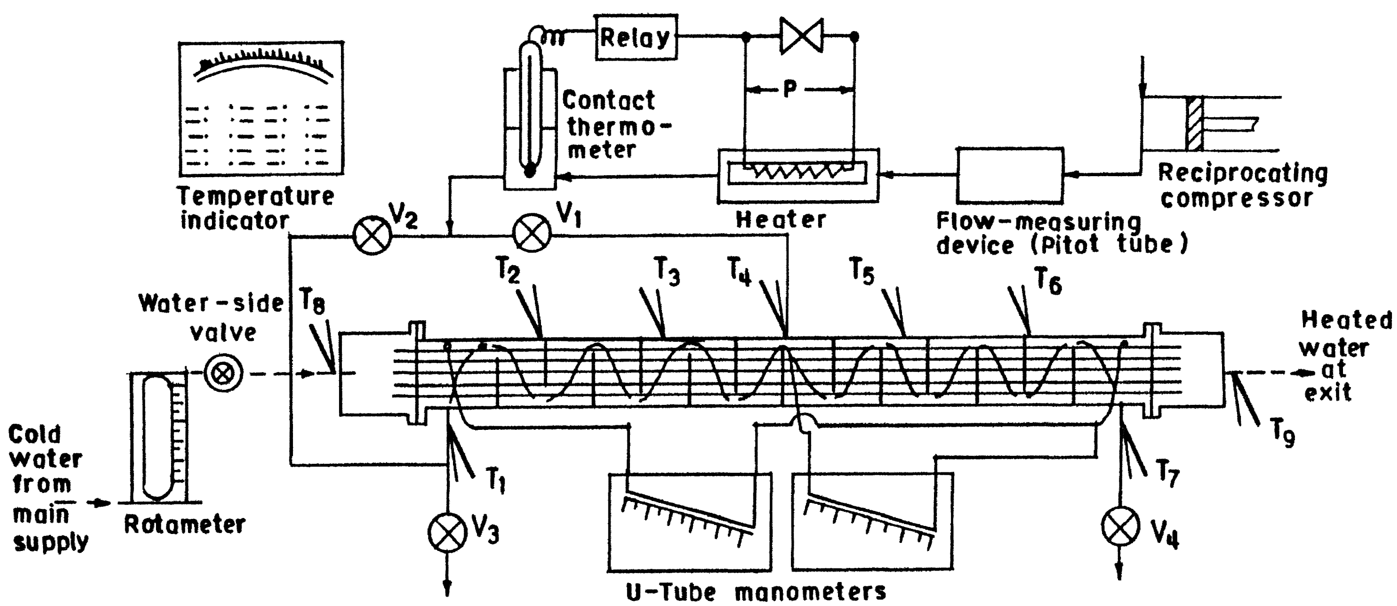
Each of these sections consists of regulating valves or relays and measuring instruments. The air flow circuit has, in addition, some accessories as the heating element and contact thermometer for temperature control.

The flow arrangement and the regulating valves are so set, that the same set-up can assume either the counter- or the split-flow configuration as per desire. The power supply to the heater is also controlled by a relay which works on switching control to maintain a pre-set temperature.

4.2 Details of Set-up and Procedure

4.2.1 Air flow circuit

(a) Compressor: The air from ambient is sucked and compressed by a reciprocating compressor to a desired pressure before being supplied to the test section.



Counterflow arrangements
Valves V_2 , V_4 - Open
V_1 , V_3 - Closed

Split flow arrangements
Valves V_1 , V_3 , V_4 - Open
V_2 - Closed

FIG. 4.1 SCHEMATIC REPRESENTATION OF EXPERIMENTAL SET-UP

(b) Flow measuring device: The supply of air was estimated by the means of a pitot tube. Its position was kept fixed inside the tube after obtaining a correction factor of 1.0037 for the value of pressure drop. From this the average velocity of flow and the volume flow rate has been calculated.

(c) Heating arrangement: Immediately after flow measurement, the air is passed over the annular heater. The heating rod is fixed inside an insulated cylindrical pipe. The power to the heater was supplied through a variac to enable the relay to cut off power supply once the desired temperature is reached.

(d) Relay and contact thermometer: The temperature of the air at the exit of the heater is sensed by a contact thermometer having a pre-set desired value. This provides an alternative path for current to flow, when this temperature is reached, so that the heater is switched off.

(e) Valves: The valve arrangement has a crucial role to play in obtaining the configuration of the exchanger. The tubing and valves have been so arranged as to get either the counter- or split-flow configuration. As shown in Figure 4.1, when solenoid valve V_1 is open and V_2 closed, the entry to the shell is at the centre and the exit at the ends leading to split flow condition; whereas if V_1 is closed and V_2 open, the entry to the shell is at one end and exit at the other, a counterflow situation.

The manually operated 12.7 mm globe valves, V_3 and V_4, are used to control the flow rate at each exit in split flow condition to ensure equal division of the air flow into the two sections of the exchanger.

4.2.2 Test exchanger: It comprises a shell, 13 baffles, 19 copper tubes of internal diameter 9.50 mm. In addition there are arrangements for measurements of pressure drops in the shell for both the configurations (Figure 4.3). The details of these components are given below:

(a) Shell: It is a flanged hollow cylinder of perspex whose internal diameter is 84 mm and thickness 3 mm. The length of the shell between the two extremes is 900 mm.

The shell has three ports: one at the centre and two on the extreme ends at diametrically opposite side. The central port is the inlet for split flow configuration and the extreme ports act as exits. In the counterflow case, the central port is closed; one of the end ports acts as the inlet and the other as the exit.

Near each of these ports, pressure tappings are made on the shell. These are connected to inclined tube manometers as shown. In the split flow configuration, the extreme ports are linked together through T-connections to the arms of another manometer in order to get equal flow on both sides. The globe valves V_3 and V_4 are adjusted carefully to get a null-point.

Thermocouples are provided at various locations to measure the temperatures at the ends and along the length of exchanger for the two fluids.

(b) Tube bundle and header: The tubes have an outer diameter of 10 mm. These are arranged in a triangular layout of pitch 15 mm. The baffles are made of 5 mm thick perspex sheet and are spaced 60 mm apart. In addition to providing a desirable flow path for shell-side fluid, they also hold the tubes in place.

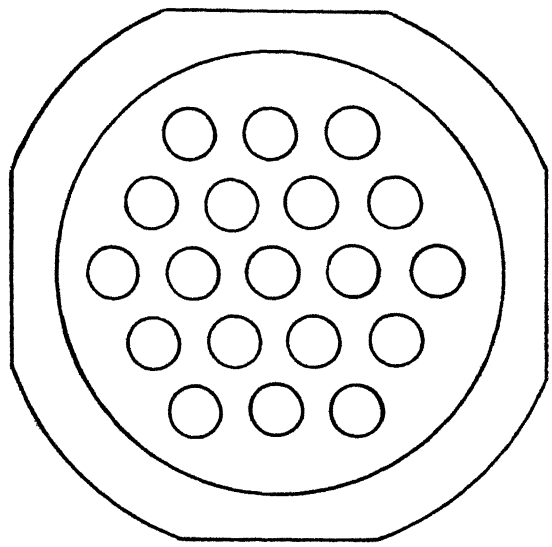


FIG. 4.2 (a) TEMPLATE FOR BAFFLES & TUBE - SHEET

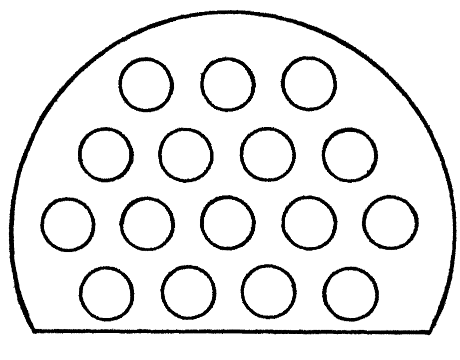


FIG. 4.2 (b) 25 % SEGMENTAL
BAFFLE

The outside diameter of the baffle plates is 83 mm and they are of the 75% segmentally cut type (Figure 4.2).

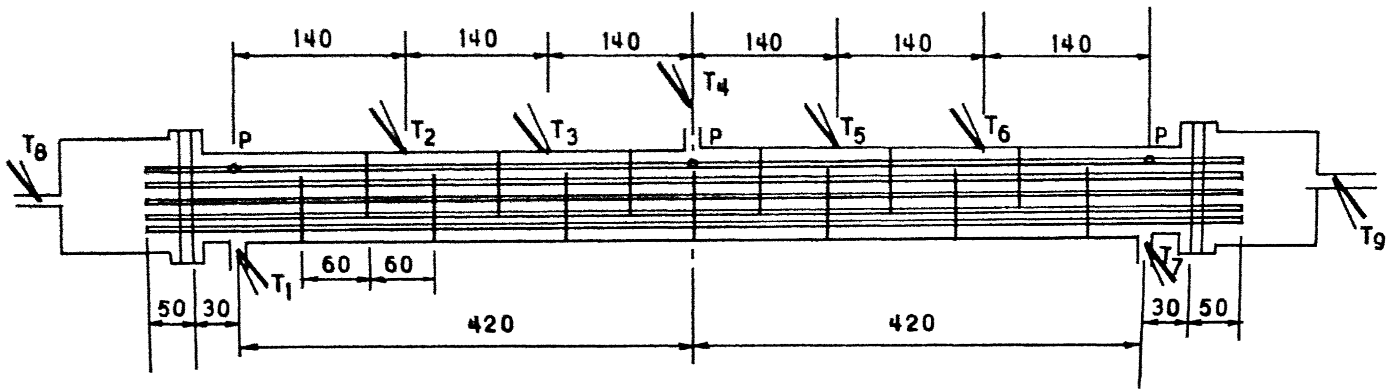
The tubes pass through a tube sheet at each end of the shell into the header for the tube-side fluid. The header itself is a cylindrical perspex block, closed at the far end except for the openings for liquids. The mating surfaces of the shell flange, tube sheet and the header flange are of good finish and held together tightly by means of bolts to prevent any leakage.

The inlet and exit ports for the tube side fluid are also provided with thermocouples for the measurement of terminal temperatures.

4.2.3 Water-flow circuit: The water path consists of the water from the main supply of water. A rotameter of capacity 9.65 litres/min. is incorporated as shown in Figure 4.1. A regulating valve is used in line to monitor the desired flow-rate. From there it flows into the inlet header and after flooding, it enters the tubes whose ends are projected slightly inside the header. After the heat exchange, the water collects in the exit header before being discharged to the drain.

4.3 Instrumentation and Measurement

4.3.1 Temperature measurement (Figure 4.3): The temperature of both the streams, air and water is measured at the entry and exit points of the shell. In addition, to estimate the variation in heat transfer and water temperature along the length of heat exchanger, the temperature of air is measured at 4 symmetric locations. The temperature of water in the tubes is obtained by heat balance.



Shell inside diameter = 84
Baffle thickness = 5
Tube o.d. = 10
T = Locations for thermocouple
P = Locations for pressure tappings

All dimensions are in mm

FIG. 4.3 LOCATIONS OF TEST-SECTION INSTRUMENTATION

The instruments used to measure the temperatures are copper-constantan thermocouples. These are connected via lead wires to a recorder which displays the temperatures on a dial directly.

4.3.2 Pressure drop measurements: To measure the pressure drop on the shell-side,an inclined tube manometer of 150 mm of water is used. A red oil with specific gravity 0.826 is used.

The pressure drop is measured across the total length of the exchanger in the counterflow configuration and across the half length in case of split flow arrangement.

4.3.3 Flow-rate measurement and control: On the water-side, the flow rate is measured by a calibrated rotameter with range from 0 to 9.65 litres/min. For the control of water flow, a 12.7 mm globe valve is provided immediately after the rotameter exit. The valve is manually operated.

On the air-side, the total flow rate is estimated by a pitot tube. The velocity profile is almost uniform, as found experimentally (the correction factor is found to be 1.0037). For the split-flow case, the division of air flow equally in the two halves is achieved by balancing the pressure values at the two exits by manipulating the globe valves V_3 and V_4.

The experiment thus records the pressure drops on the shell-side for the two configurations under investigation; the temperature variation along the length of the exchanger, and the flow rates of the fluids at which these measurements are made.

CHAPTER - 5

RESULTS AND DISCUSSIONS

5.1 Numerical Results

The flow chart of the computer program has been given in Appendix 7 for the comprehensive design of heat exchanger for the counter-flow configuration and subsequent split-flow configuration.

At first, the program designs a counter-flow exchanger for a given heat duty and inlet conditions of the fluids involved. Thereafter, the design of split-flow configuration is undertaken for the same heat duty by changing

(i) number of tubes

or (ii) the length of tubes.

The program is based on the commercially available sizes of tubes, shells, tube layouts (square or triangular positions), tube bundle data and on correlations for heat transfer coefficients, friction factors, property values of fluids etc. The prediction of U value is based on the approach followed by Kern [3].

The computed results show that if a counter-flow exchanger is converted into the split-flow, there is reduction in heat duty by 6-10% for the ranges of shell-side Reynolds number. This reduction is brought about by the decrease in overall heat transfer coefficient U, because the individual film coefficients decrease with reduction in the Reynolds number of the flow. In fact, the fall in the value of U is even greater. But the true temperature difference in split-flow is greater than that in

counterflow because the temperature difference along a major portion of the length and at the exit of the split-flow exchanger is more.

The ratio of the shell-side pressure drop varies in a narrow range of 0.130-0.133 for air-water exchangers and 0.1423-0.1427 for water-water and ethyl alcohol-water exchangers for a wide practical range of parameters. The ratio remains fixed for most of the cases, because the slope of the friction factor versus Reynolds number curve is constant on a log-log plot, in the range of practical shell-side values. Therefore, the ratio of the friction factor values on dividing the flow rate remains more or less fixed. The minor variations are due to the change in fluid properties due to change in the mean temperature values, etc.

The computer program also designs alternative split-flow configurations where attempts are made to compensate for the reduction in heat-duty either by increasing the length of the exchanger or by increasing the number of tubes in the bundle.

In the case of increase in length, the compensation for the heat-duty is assured and the pressure-drop on the shell-side may drop sometimes. This is due to a re-adjustment of baffle spacing, which might result in a reduction of the Reynolds number of flow. Usually, the pressure drop tends to increase.

When the compensation is attempted by increasing the number of tubes, the design selects the next higher number of tubes. In general, the greater number of tubes render the higher heat duty exchanger as the tubes are selected in discrete steps.

Neither of these alternative approaches to design are distinctly superior. Depending upon a particular set of desired specifications, any one of them might lead to a more suitable design.

Figure A.10 gives the temperature profiles for both configurations of the same exchanger. The counterflow yields higher temperature rise as compared to split-flow.

5.2 Experimental Results

The shell-side pressure drop variation for the counter- and split-flow configurations has been taken for varying flow rates of the shell-side fluid (air). The results are reported in Table 5.1 for the various values of shell-side Reynolds number. Figures 5.1 and 5.2 show the variation in the values of pressure drops and the ratio, respectively.

It is found that the experimental value for the pressure drop ratio is quite high as compared to the various theoretically predicted values. This might be due to the presence of the orifice effects in the clearances between the baffle and the shell-wall and the tubes and the tube-holes in the baffles.

The computed value of the pressure-drop ratio is in close agreement with the results obtained by using the theoretical expressions [12]:

$$f = \frac{0.316}{Re^{0.25}} = 0.316\left(\frac{}{G_s D_e}\right)^{0.25} \tag{5.1}$$

and

$$\frac{\Delta p_1}{\Delta p_2} = \frac{(f\ G_s^2\ L)}{(f\ G_s^2\ L)_2} = \frac{(G_s/2)^{1.75}\ L/2}{G_s^{1.75}\ L} = 2^{-2.75} = 0.149 \tag{5.2}$$

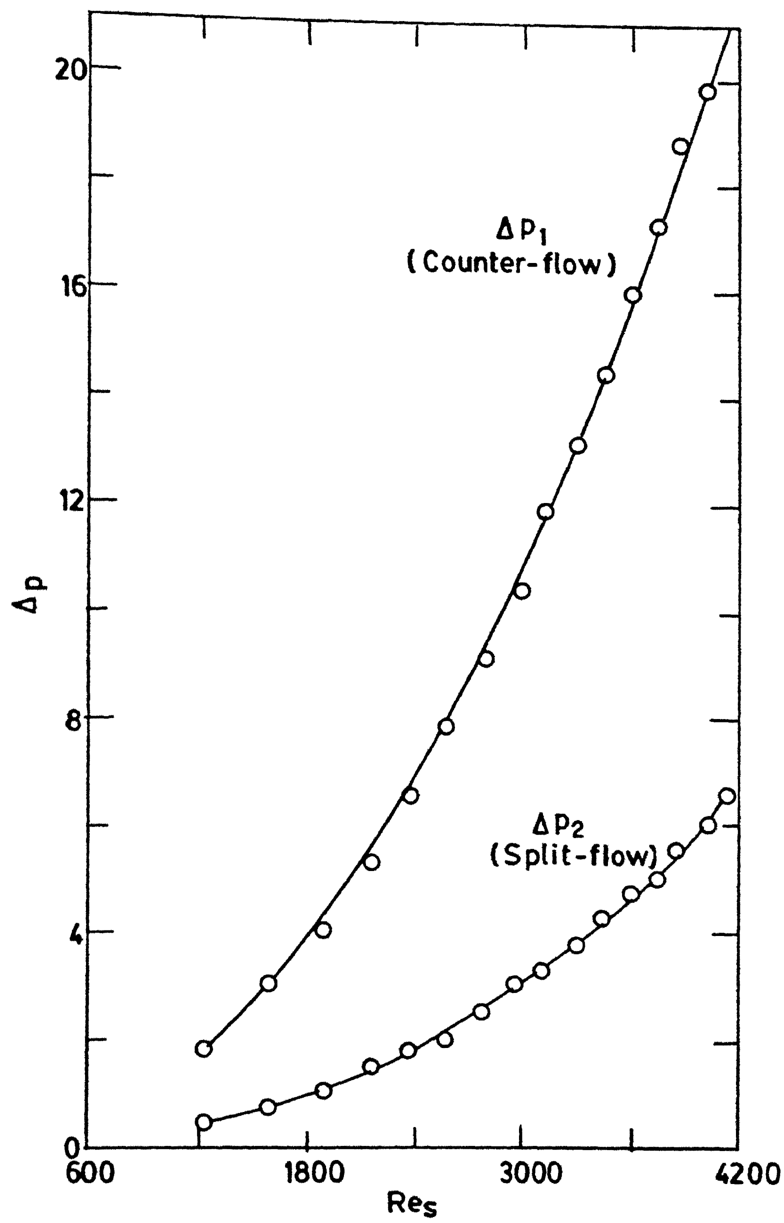


FIG. 5·1 VARIATION OF SHELL-SIDE PRESSURE DROP WITH SHELL-SIDE REYNOLDS NUMBER

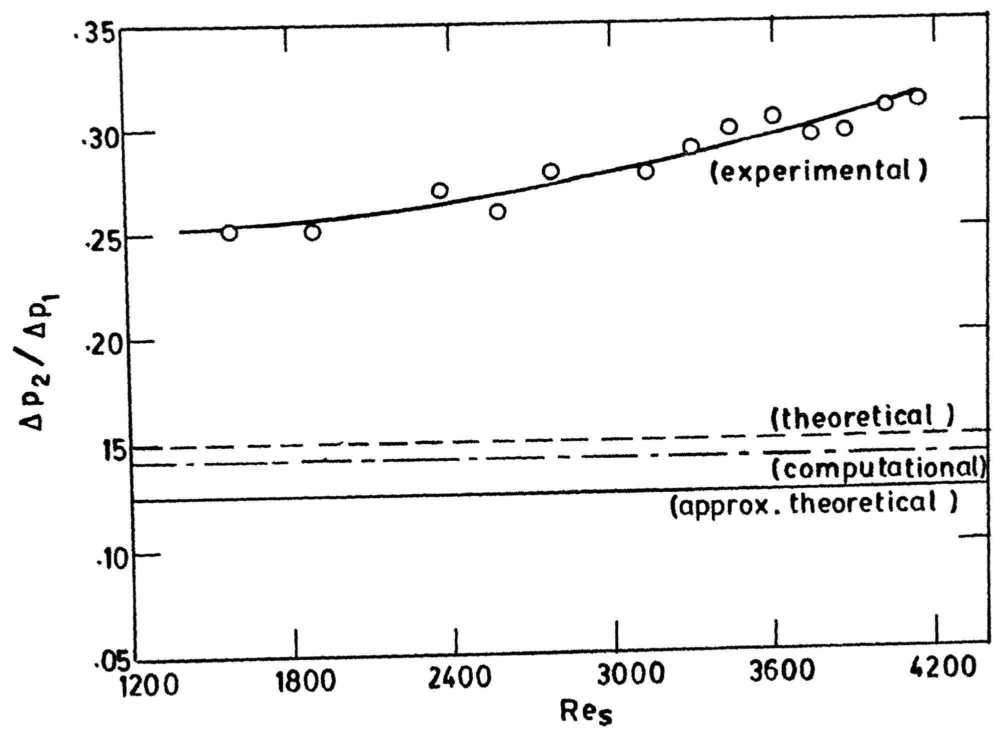


FIG. 5.2 VARIATION OF PRESSURE RATIO WITH SHELL-SIDE REYNOLDS NUMBER

Table 5.1

Pressure drops in counter- and split-flow configurations

S. No.	Pitot tube pressure drop Δh (mm of water) (MEASURED)	Re_s = $202.32 \times (\Delta h)^{\frac{1}{2}}$ (CALCULATED)	Δp_1 (mm of water) (MEASURED)	Δp_2 (mm of water) (MEASURED)	$\frac{\Delta p_2}{\Delta p_1}$
1	36.4	1220	1.778	0.508	0.2857
2	61.8	1590	3.048	0.762	0.2500
3	87.2	1890	4.064	1.016	0.2500
4	112.6	2146	5.334	1.524	0.2857
5	138.0	2377	6.604	1.778	0.2692
6	163.4	2586	7.874	2.032	0.2581
7	188.8	2780	9.144	2.540	0.2778
8	214.2	2961	10.414	3.048	0.2927
9	239.6	3132	11.938	3.302	0.2766
10	265.0	3294	13.208	3.810	0.2885
11	290.4	3448	14.478	4.318	0.2982
12	315.8	3595	16.002	4.826	0.3016
13	341.2	3737	17.272	5.080	0.2941
14	366.6	3875	18.796	5.588	0.2973
15	392.2	4005	19.812	6.096	0.3076
16	417.4	4134	21.336	6.604	0.3095

for water and ethyl alcohol. The same result is shown in Figure 5.2. But it is substantially different for air. This could be due to the effect of the variation in the density with temperature (which is larger in the case of air as compared to the other two fluids) which has been neglected in [12].

The variation of temperature in the split-flow arrangement has been studied and the results are reported in Table 5.2.

The flow rate values for the two fluids were

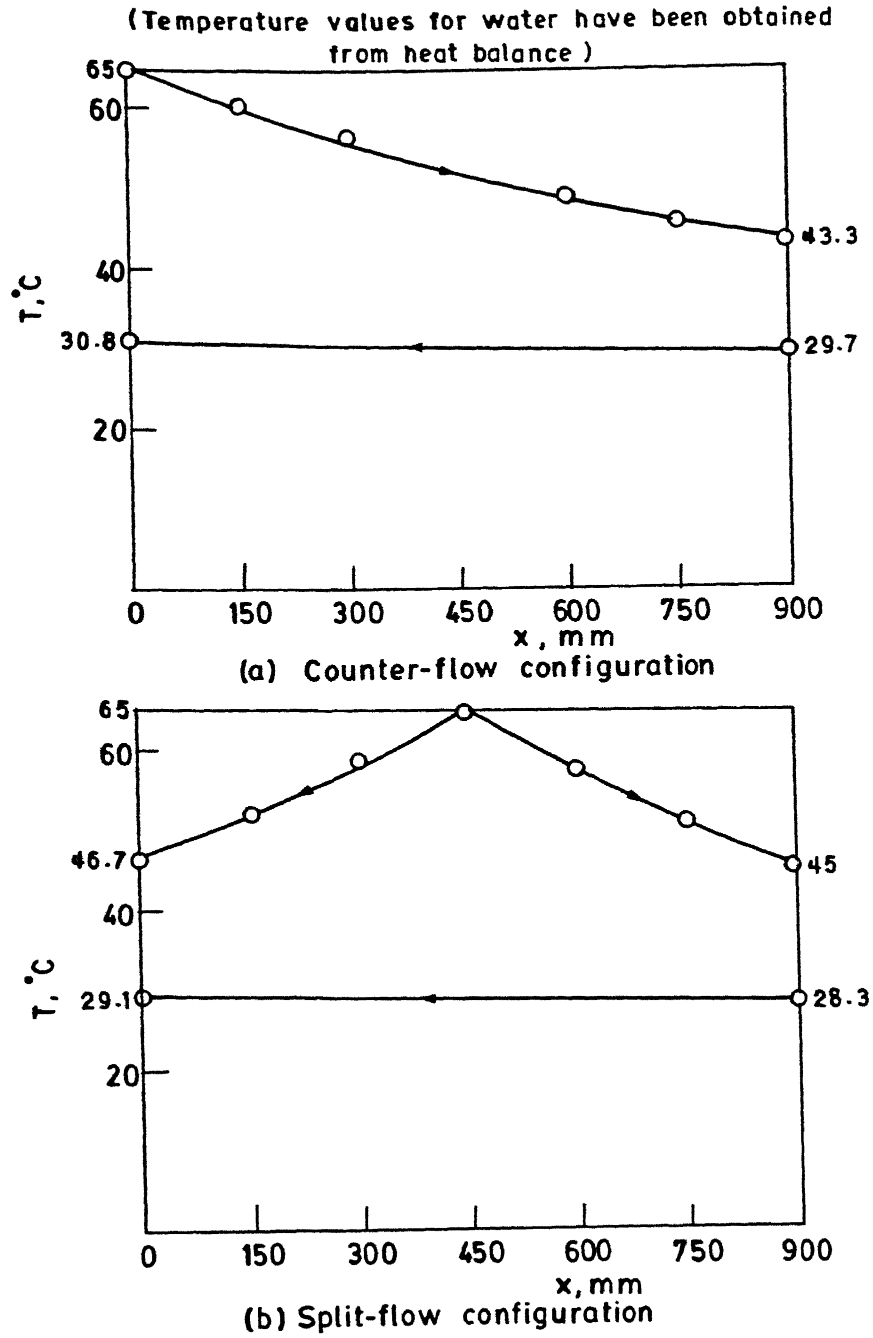
Air	-	5.08 lit/s
Water	-	3.1545 lit/min.

The variation in the heat duty for various inlet air conditions are also given at the end of the Table 5.2. The heat duty decreases by 10-15% due to split-flow configuration for the same exchanger. A typical temperature profile is shown in Figure 5.3.

Table 5.2

Temperature variation in counter- and split-flow configurations

Thermo-couple No.	SET I Temperature (C)		SET II Temperature (C)		SET III Temperature (C)		SET IV Temperature (C)	
	Counter-flow	Split-flow	Counter-flow	Split-flow	Counter-flow	Split-flow	Counter-flow	Split-flow
1	46.1	39.2	51.7	41.1	57.2	42.8	65.0	46.7
2	44.2	41.4	49.2	44.4	53.4	47.2	60.6	52.2
3	42.5	43.6	46.9	48.1	50.8	51.9	56.7	58.6
4	-	46.4	-	51.7	-	56.9	-	65.0
5	39.4	43.3	42.8	47.2	45.0	51.9	49.4	57.8
6	38.1	40.3	41.1	43.6	42.8	46.7	46.4	51.1
7	36.7	37.2	39.2	40.0	40.6	41.7	43.3	45.0
8	29.4	27.8	29.4	27.8	29.7	28.1	29.7	28.3
9	30.0	28.1	30.0	28.1	30.6	28.6	30.8	29.1
Percentage reduction in heat duty for same exchanger	14.71		12.5		10.0		11.54	



(a) Counter-flow configuration

(b) Split-flow configuration

FIG. 5.3 EXPERIMENTAL VARIATION OF TEMPERATURE ALONG THE LENGTH OF EXCHANGER

CHAPTER - 6

CONCLUSIONS AND SUGGESTIONS

6.1 Conclusions

1. A comprehensive computer program has been developed for the counter- and split-flow configurations based on practical data. The program encompasses various possibilities of design considerations to meet a particular heat duty requirement.
2. The ratio of the shell-side pressure-drop in split-flow arrangement to that in counter-flow arrangement is found out numerically as 0.130-0.133 for air on shell-side and 0.1423-0.1427 for water and ethyl alcohol. These are quite close to reported results.
3. The experimental values for this ratio have been obtained as 0.25-0.31. Thus, one should keep this fact in mind while opting for the split flow configuration in place of counter-flow exchanger.
4. The computed heat duty in case of split-flow arrangement is reduced by about 6-10%. However the experimental values give a deviation of 10-15%.
5. The effect of compensating this loss in heat duty upon the pressure drop may be favourable or not, depending upon the method of compensation chosen. Neither of the two methods — increasing the number of tubes with the length fixed or

increasing the length of tubes with their number remaining constant – is inherently better. Their relative superiority is dependent upon the desired specifications.

6.2 Suggestions

1. The design algorithm and program can be extended to take into consideration fluids other than air, water and ethyl alcohol considered in this study.
2. For taking into consideration the effect of temperature upon the properties of the fluids, where the variations are significant, the mean fluid temperature should be evaluated as given in Appendix 2 and the viscosity effects should be incorporated as in Appendix 3.
3. For thick walled tubes, the conduction resistance should be accounted for, as per the details mentioned in Appendix 1.
4. The experiment could be extended for larger capacity heat exchangers along with higher temperature ranges and phase change cases.

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APPENDIX 1

OVERALL COEFFICIENT OF HEAT TRANSFER

The resistance to the transfer of heat across a pipe-wall consists of the pipe-side fluid film, the pipe-wall and the outside film. The overall heat transfer coefficient is defined as the inverse of total resistance i.e.

$$U = \frac{1}{\Sigma R} \tag{A.1}$$

If the area based on the outside diameter is taken as the reference area, the expression for the overall heat transfer coefficient is:

$$U = \frac{1}{\dfrac{1}{h_i\left(\dfrac{A_i}{A_o}\right)} + \dfrac{D_o}{2k}\ln\left(\dfrac{D_o}{D_i}\right) + \dfrac{1}{h_o}} \tag{A.2}$$

If $h_i\left(\frac{A_i}{A_o}\right)$ is replaced by h_{io}; and the tube material has a high value of k and $\frac{D_o}{D_i}$ close to 1 (i.e. small thickness), so that the conductive resistance is small, the expression simplifies to:

$$U = \frac{h_{io}\, h_o}{h_o + h_{io}} \tag{A.3}$$

APPENDIX 2

VISCOSITY EFFECTS IN HEAT TRANSFER

For pipe-flow with heating, the viscosity near the wall is lower than the bulk of the fluid. This leads to higher velocity and heat transfer coefficient on the inside. The reverse occurs in case the fluid is getting cooled. Similar effects are observed on the outside of the tube as well.

The effect of these is accounted by the well known modified correlation of Sieder and Tate:

$$Nu = \alpha\,(Re)^p\,(Pr)^q\,\left(\frac{\mu}{\mu_w}\right)^r \tag{A.4}$$

where μ_w is evaluated at the wall temperature t_w.

The expressions for t_w are:

$$t_w = t_c + \frac{h_o}{h_{io} + h_o}(T_c - t_c)$$

(cold fluid inside the tube)

(A.5)

$$= t_c + \frac{h_{io}}{h_{io} + h_o}(T_c - t_c)$$

(hot fluid inside the tube)

The value of r is found to be 0.14. For fluids with μ weak dependence on temperature (e.g. water, air), the viscosity factor $\frac{\mu}{\mu_w} = 1$.

APPENDIX 3

THE AVERAGE FLUID TEMPERATURE

In any exchanger, the dependence of fluid properties upon temperature implies that the individual film coefficients h_i, h_o and hence the overall coefficient U vary along the length of exchanger. This variation in U can be accounted either by numerical integration of the equation of heat balance

$$dQ = U \, dA \, \Delta t \tag{A.7}$$

or by assuming a linear dependence of U on the fluid temperature. The former approach is more accurate, but more time consuming and often the incremental accuracy is not commensurate to the effort.

The latter approach is due to Colburn. Instead of taking the LMTD as the true temperature difference in a case with varying U, it assumes U to be a linear function of temperature. The temperature at which the value of U should be evaluated is derived as below. If

$$U = a(1 + bt) \tag{A.8}$$

the integration of the heat balance equation (A.7) leads to

$$\frac{Q}{A} = \frac{U_1 \, \Delta t_2 - U_2 \, \Delta t_1}{\ln\left(\frac{U_1 \, \Delta t_2}{U_2 \, \Delta t_1}\right)} \tag{A.9}$$

where subscripts 1 and 2 refer to cold and hot junction values.

The Eq. (A.9) reduces to the form of LMTD equation with a constant value U_{av} gives

$$\frac{Q}{A} = U_{av} \frac{\Delta t_2 - \Delta t_1}{\ln(\frac{\Delta t_2}{\Delta t_1})} \tag{A.10}$$

If t_{av} is the average temperature, from Eqs. (A.9) and (A.10), it can be evaluated as

$$\begin{aligned} t_{av} &= t_1 + F(t_2 - t_1) \\ T_{av} &= T_2 + F(T_1 - T_2) \end{aligned} \tag{A.11}$$

where F is the Caloric fraction of the controlling stream (i.e. the one with lower value of h) defined as

$$F = \frac{(t_{av} - t_1)}{(t_2 - t_1)} = \frac{(1/K) + r/(r-1)}{1 + \frac{\ln(K+1)}{\ln r}} - \frac{1}{K} \tag{A.12}$$

and $\quad K = \dfrac{U_2 - U_1}{U_1}$; $\quad r = \dfrac{\Delta t_1}{\Delta t_2}$

For cases where $U_2 \simeq U_1$ or properties weakly dependent on temperature $F \rightarrow 0.5$ so that the average temperature can be taken to be the mean temperature of the fluid. Such is the case with air and water.

APPENDIX 4

TRUE TEMPERATURE DIFFERENCE FOR SPLIT-FLOW

In the split-flow arrangement, one half behaves as if in counterflow and the other as if in parallel-flow. If the total flow rates are m and M on the tube- and the shell-side respectively, and $R = \frac{mc}{MC}$, then in the counterflow section (refer to Figure 3.2(b)), heat balance over an element gives:

$$dq = mc\ dt = \frac{MC}{2}\ dT = U(T - t)dA \qquad (A.13)$$

On rearrangement it can be modified to

$$\frac{d(T - t)}{(T - t)} = \frac{U(2R - 1)}{mc}\ dA \qquad (A.14)$$

The integration of this equation from 0 to x gives the temperature difference at any location x in the counterflow section as

$$\frac{(T_x - t_x)}{(T_2' - t_1)} = e^{\frac{UA_x}{mc}(2R-1)} \qquad (A.15)$$

The value of t_x, from overall heat balance upto x

$$\frac{MC}{2}\ (T_x - T_2') = mc(t_x - t_1) \qquad (A.16)$$

can be substituted in Eq. (A.15) to obtain

$$\frac{(T_x - t_1)}{(T_2' - t_1)} = \frac{1}{(1 - 2R)} - \frac{2R}{(1 - 2R)}\ e^{\frac{UA_x}{mc}(2R-1)} \qquad (A.17)$$

as the temperature variation along x.

A similar exercise for parallel flow section give:

$$dq = mc\ dt = -\frac{MC}{2}\ dT = U(T - t)dA \qquad (A.18)$$

APPENDIX 5

PROPERTIES AS FUNCTIONS OF TEMPERATURE

(All values are in SI units, T in C)

A.5.1 Coefficient of viscosity μ:

$$\mu_a = 1.48883 \times 10^{-6} (T + 273)^{1.5}/(T + 393)$$

$$\mu_v = 0.6288959/(44.3002 + T)^{1.54579} \qquad 0 < T < 100$$

$$\mu_{et} = 8.20 \times 10^{6}/(200 + T)^{4.2} \qquad 0 < T < 75$$

A.5.2 Thermal conductivity k:

$$k_a = 1.9676 \times 10^{-3}(T + 273)^{1.5}/(T + 398)$$

$$k_v = 1.47016 \times 10^{-3}\ T + 0.57077 \qquad 0 < T < 100$$

$$k_{et} = 0.182[1 - 0.00061(T - 20)] \qquad 0 < T < 80$$

A.5.3 Specific heat C:

$$C_a = 1004$$

$$C_v = 4225.06 - 2.002\ T \qquad T < 15$$

$$= 4195.03 - 1.402(T - 15) + 0.04(t - 15)^2 \qquad 15 < T < 40$$

$$= 4185.02 - 0.3(T - 40) \qquad T > 40$$

$$C_{et} = 2240 + 5.4\ T + 0.088\ T^2 \qquad 0 < T < 70$$

A.5.4 Density S: $t = (T/100)$

$$S_a = 1.293 - 0.451\ t - 0.14\ t^2 + 0.6\ t^3$$

$$S_v = 1000 + 2.1663\ t - 570.33\ t^2 + 0.033\ t^3 - 10.417\ t^4$$

$$S_{et} = 803.1 - 0.5267\ T$$

APPENDIX 6

CORRELATIONS FOR HEAT TRANSFER AND FRICTION FACTOR

A.6.1 Heat transfer:

(a) Tube-side (valid for Re 8000)

$$j_H = 0.027\ Re^{0.8} \quad \text{for air}$$

$$h_i = F(3178.86 + 46.36\ T)\ V^{0.80214} \quad \text{for water}$$

where $F = 1.08 - 0.1864\ \ln(d_i/1.016)$

(b) Shell-side

$$j_H = 0.36\ Re^{0.55} \qquad Re > 2000$$

$$= 0.63258\ Re^{0.46107}/e^{-(4.87716 \times 10^{-5}\ Re)} \qquad Re < 2000$$

A.6.2 Friction factor:

(a) Tube-side:

$$f = \frac{16}{Re} \qquad Re < 2300$$

$$= 0.00695 + 5.84285 \times 10^{-6}(Re - 2300) \text{ for tubes}$$

$$= 0.00695 + 8.14285 \times 10^{-6}(Re - 2300) \text{ for pipes}$$

$$2300 < Re < 3000$$

$$= 0.0014 + 0.125\ Re^{-0.32} \quad \text{for tubes}$$

$$Re > 3000$$

$$= 0.035 + 0.264\ Re^{-0.42} \quad \text{for pipes}$$

(b) Shell-side:

$$f = 0.64335 \exp(0.06066\ Re^{0.67083})\ /Re^{1.29342}$$

$$Re < 50$$

$$= \quad 0.011054 \exp(20.18008\ Re^{-0.94295})/Re^{0.18536} \qquad 50 < Re < 2000$$

$$= \quad 0.00087(10^6/Re)^{0.190934} \qquad Re > 2000$$

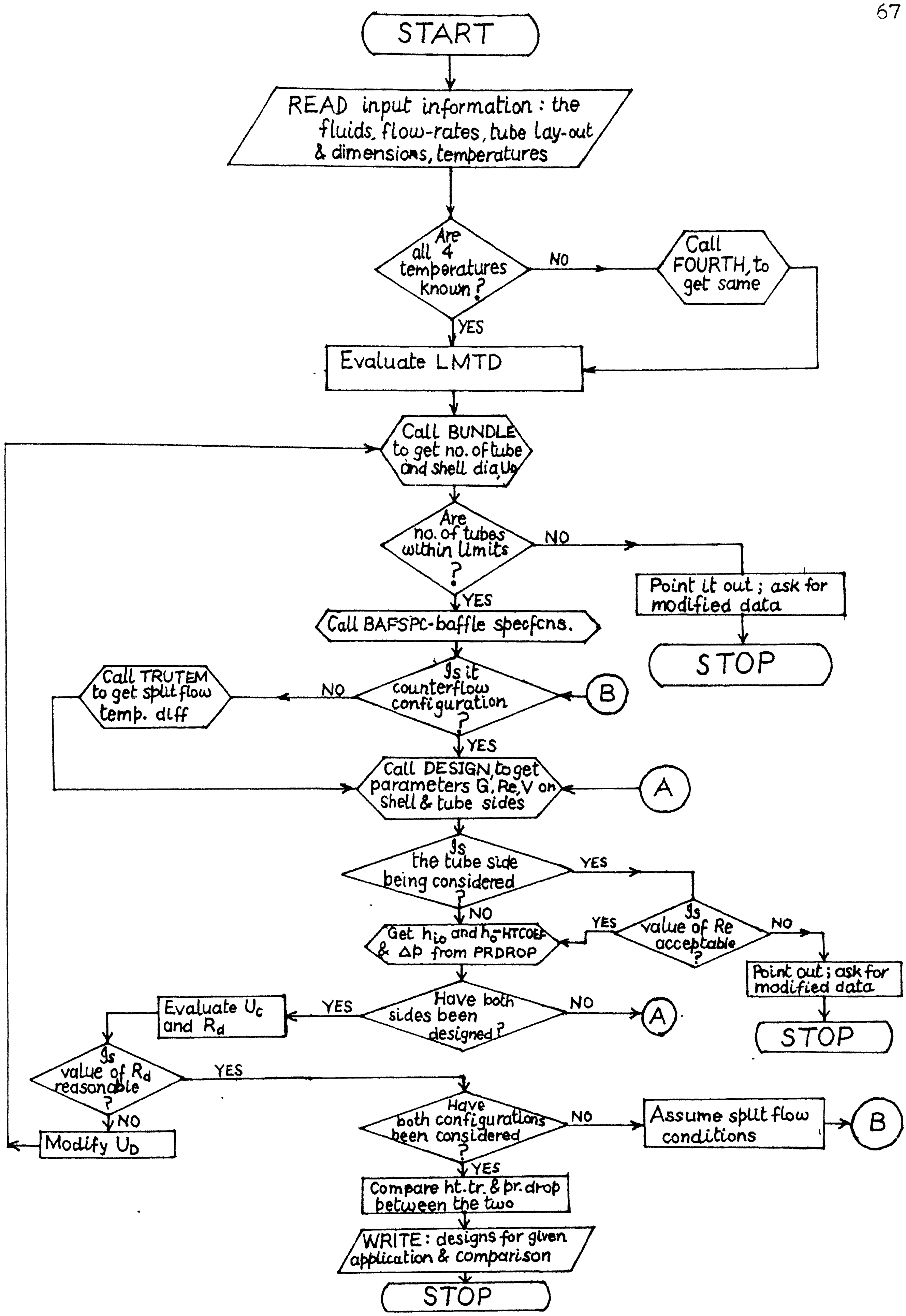
APPENDIX 7

THE COMPUTER PROGRAM AND A SAMPLE SOLUTION

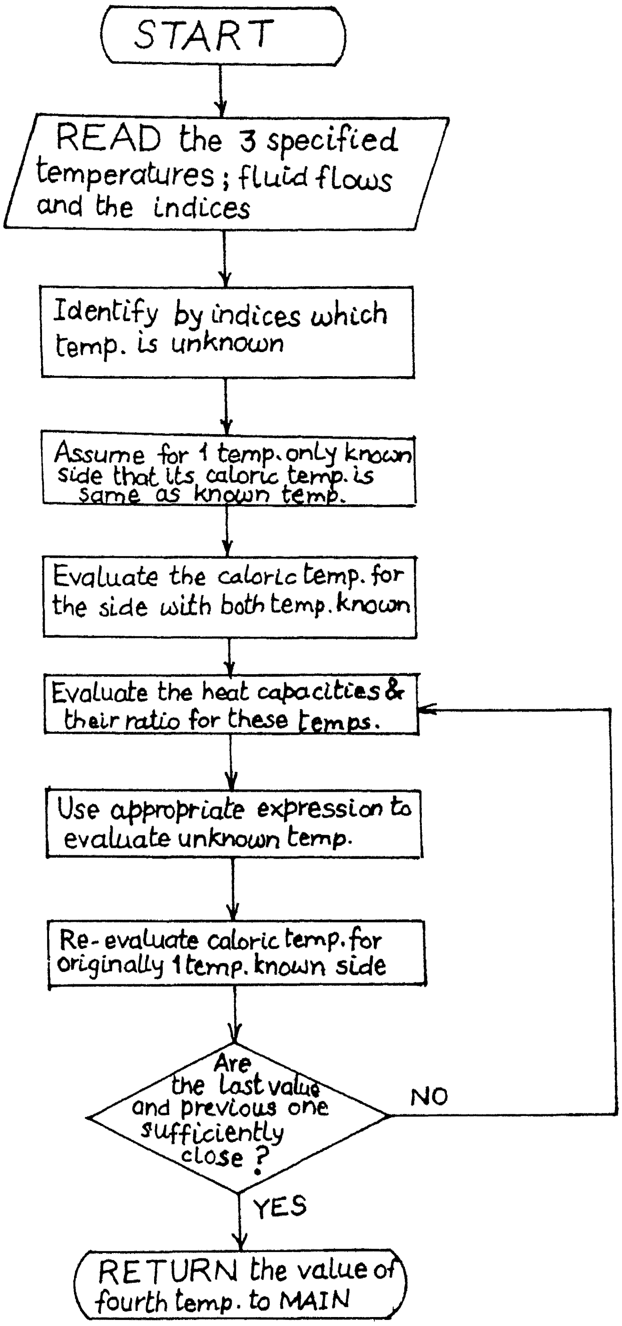
The working of the program has been explained by means of flow-chart in Figures A.1 to A.9. The section also contains the solution to a sample problem. The input data needed for the program are explained below briefly.

IFLSH, IFLTB	-	index values for the shell-side and tube-side values, 1 signifying water, 2 air and 3 ethanol
SHMF, TBMF	-	the respective flow rates for fluids
LAY, IOD, IBWG	-	indices designating the chosen layout, outside diameter and BWG value for tubes
TBL	-	tube length
IT(I), I=1,4	-	indices having value either 0 or 1. A value zero signifies that particular temperature has not been specified
THT, THO(1), TCI, TCO(1)	-	the terminal temperatures on the hot and the cold sides respectively for counterflow
IP	-	a value 1 signifies the cold fluid flows in a pipe, else in a tube
IBAF, SPCB	-	a value 1 for IBAF implies that approximate baffle spacing desired (SPCB) is provided
IXMAX	-	number of points at which the temperature values are calculated in split-flow.

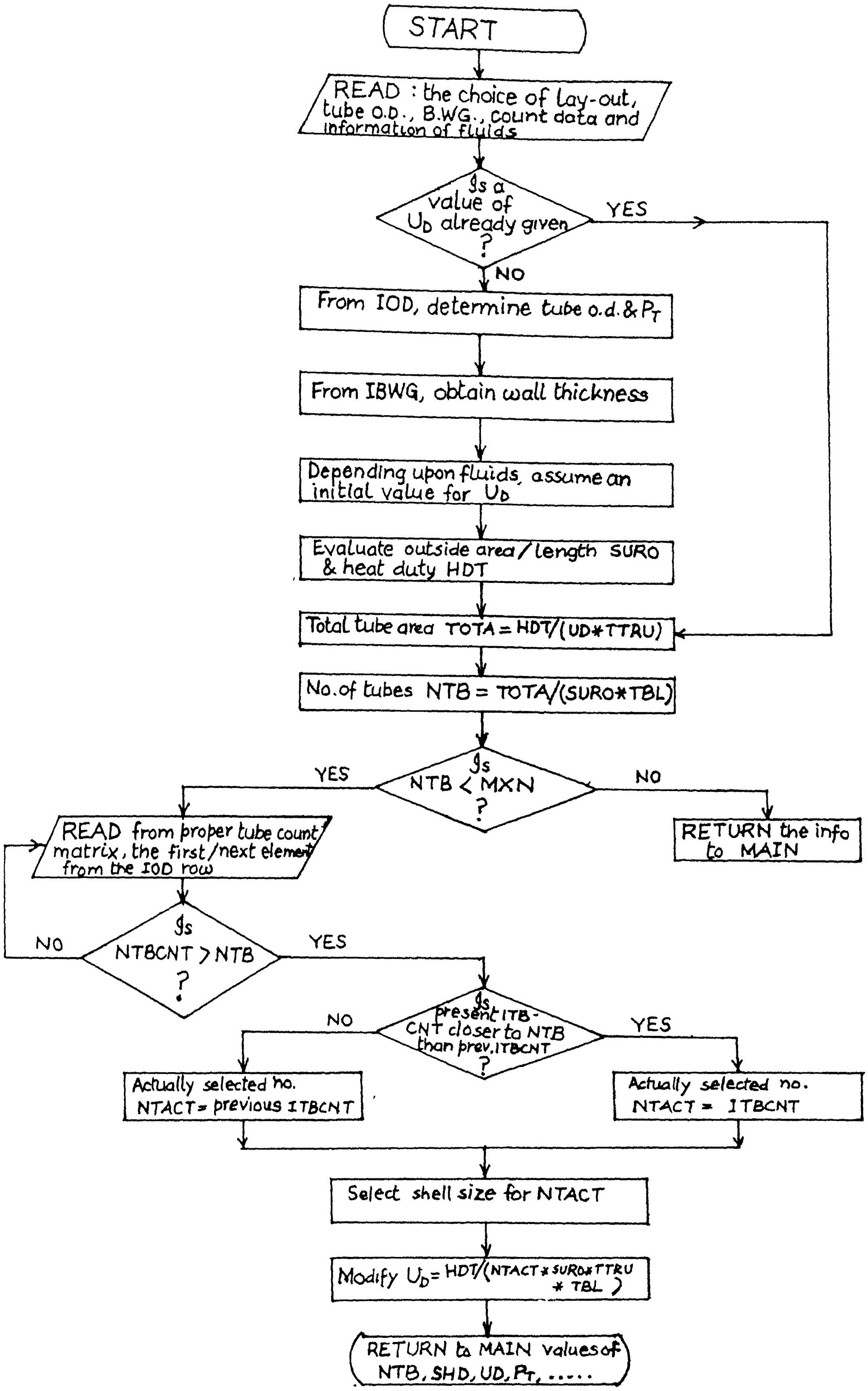
Figure A.10 shows the temperature profiles as obtained from the sample solution for the two configurations.



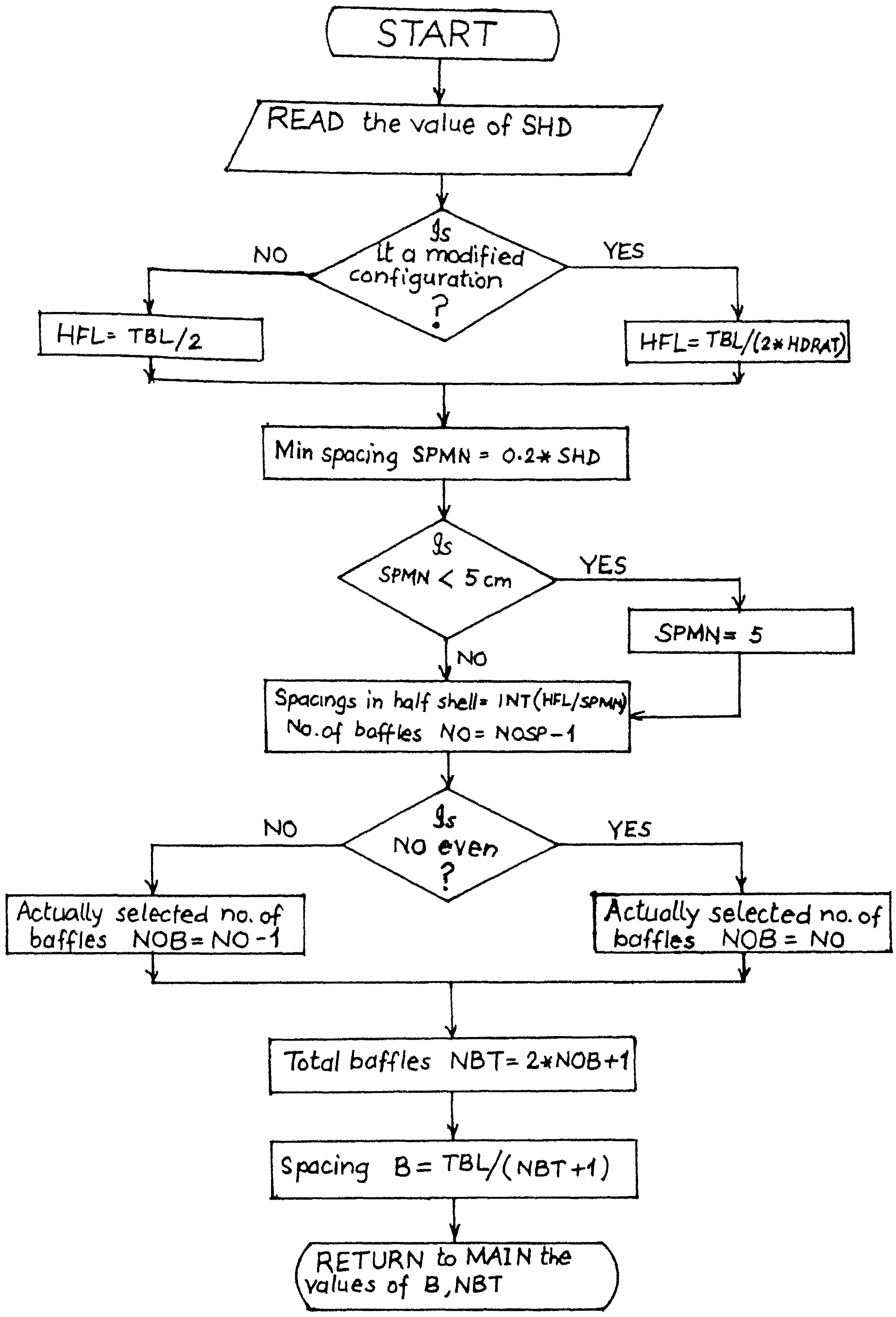
A-1 : MAIN PROGRAM



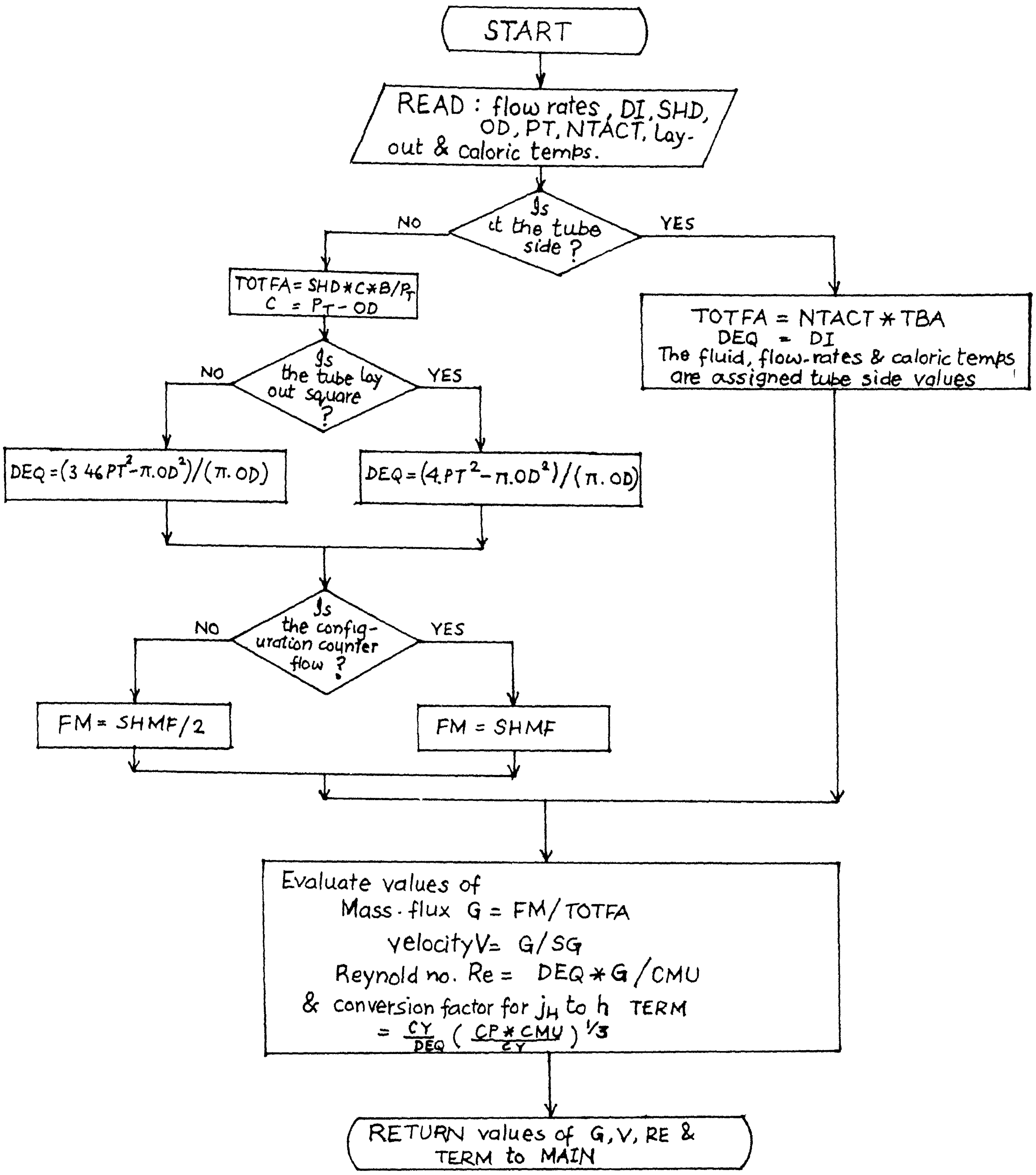
A-2 : SUBROUTINE FOURTH



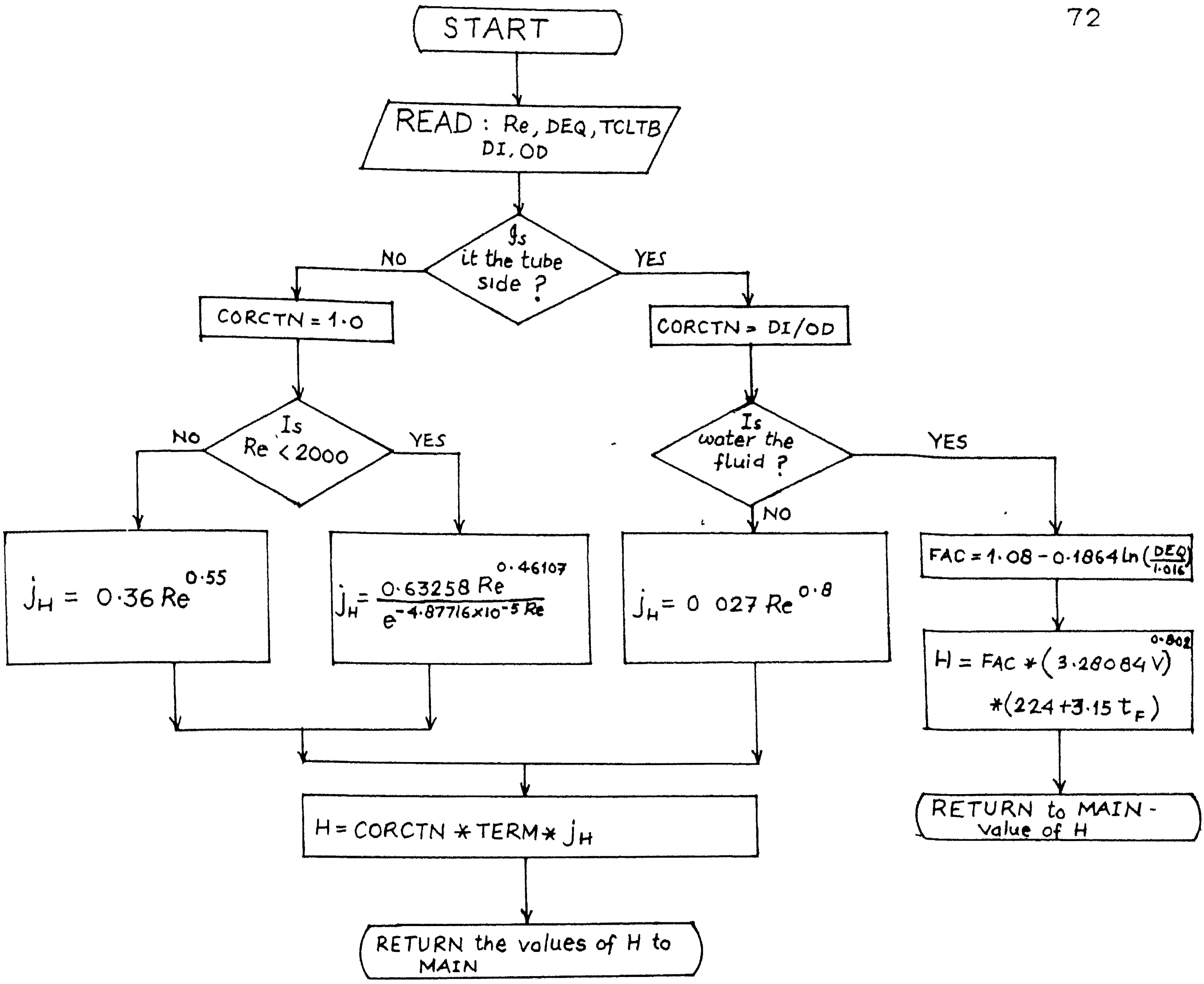
A-3 : SUBROUTINE BUNDLE



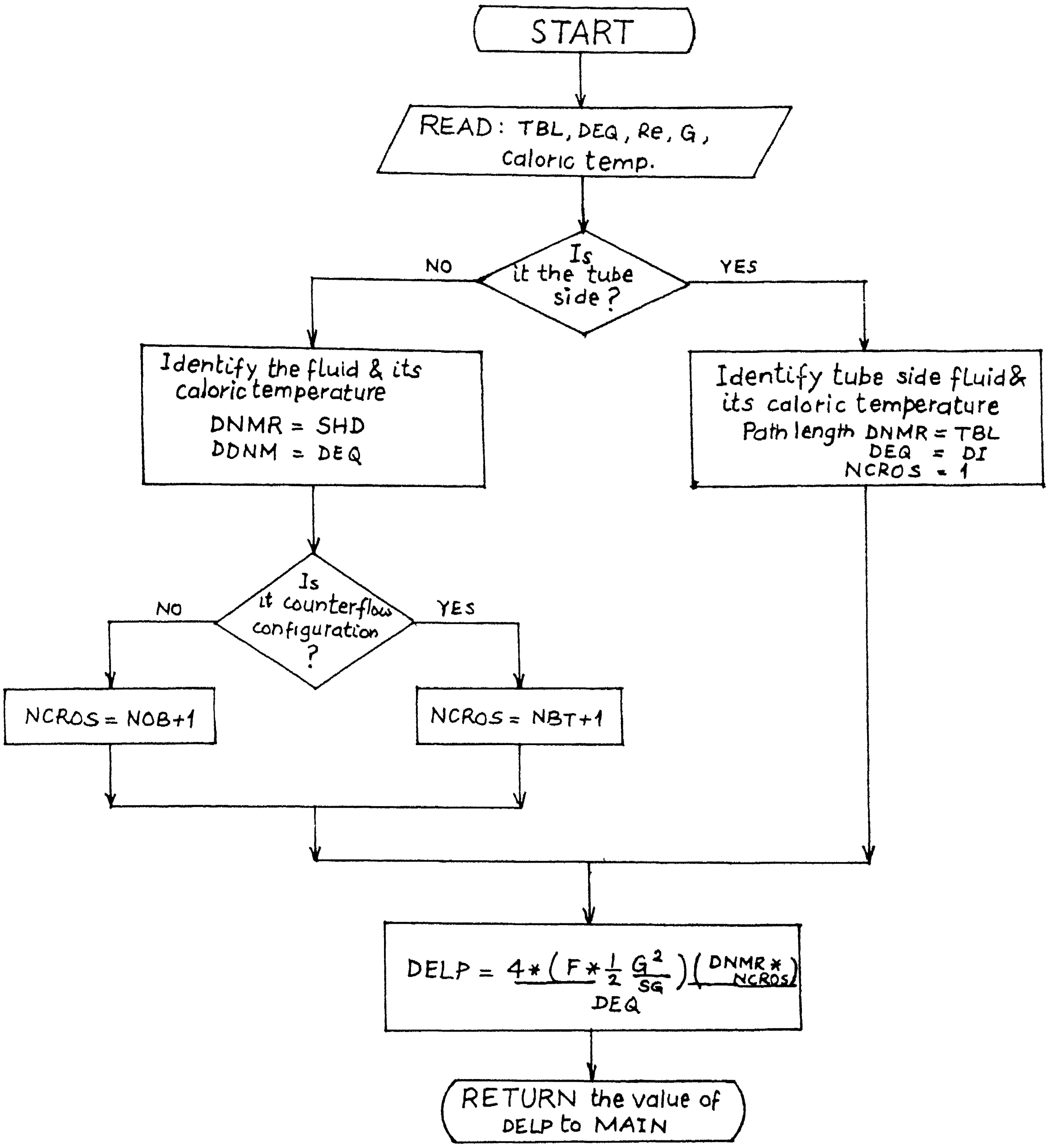
A - 4: SUBROUTINE BAFSPC



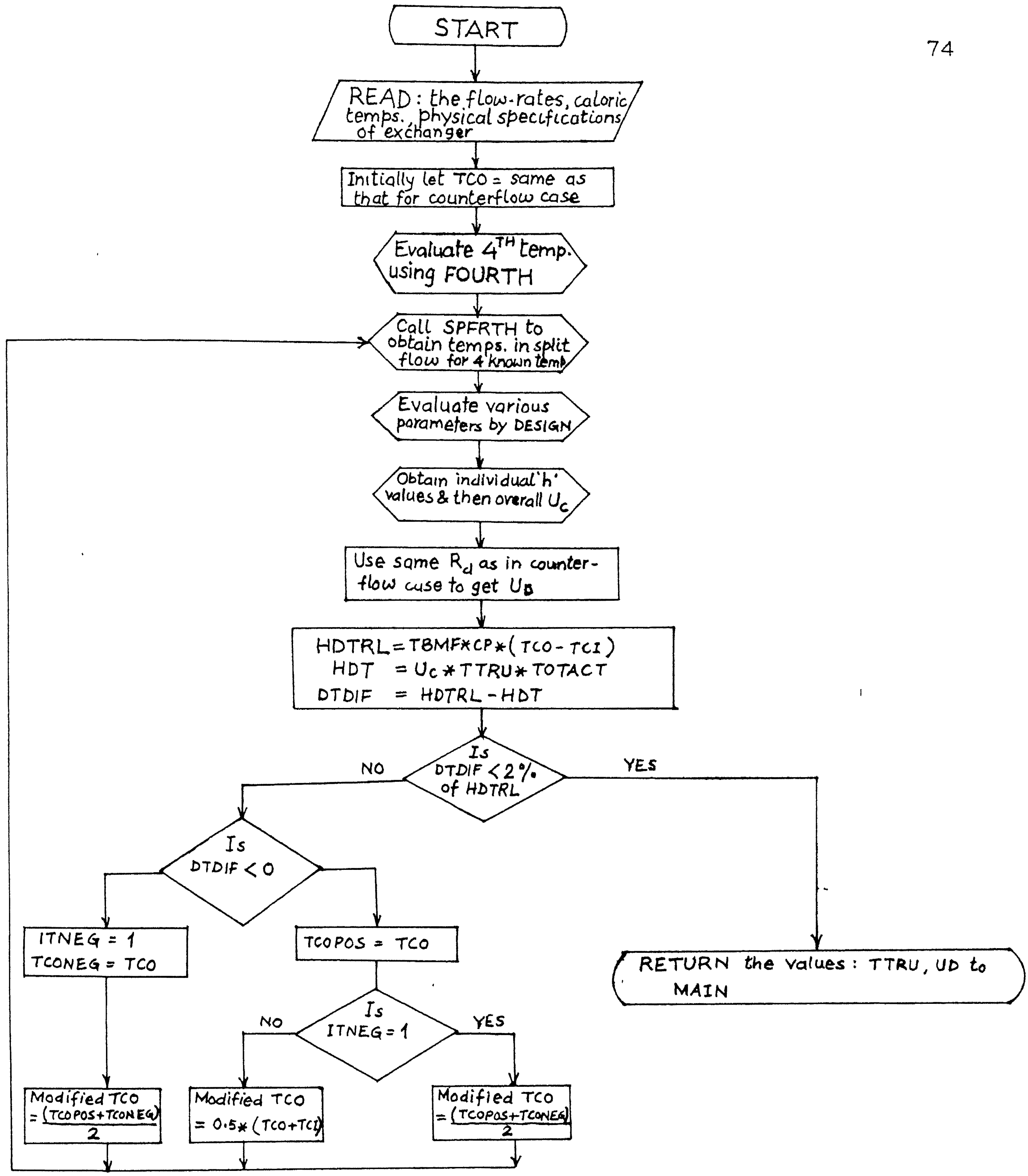
A - 5: SUBROUTINE DESIGN



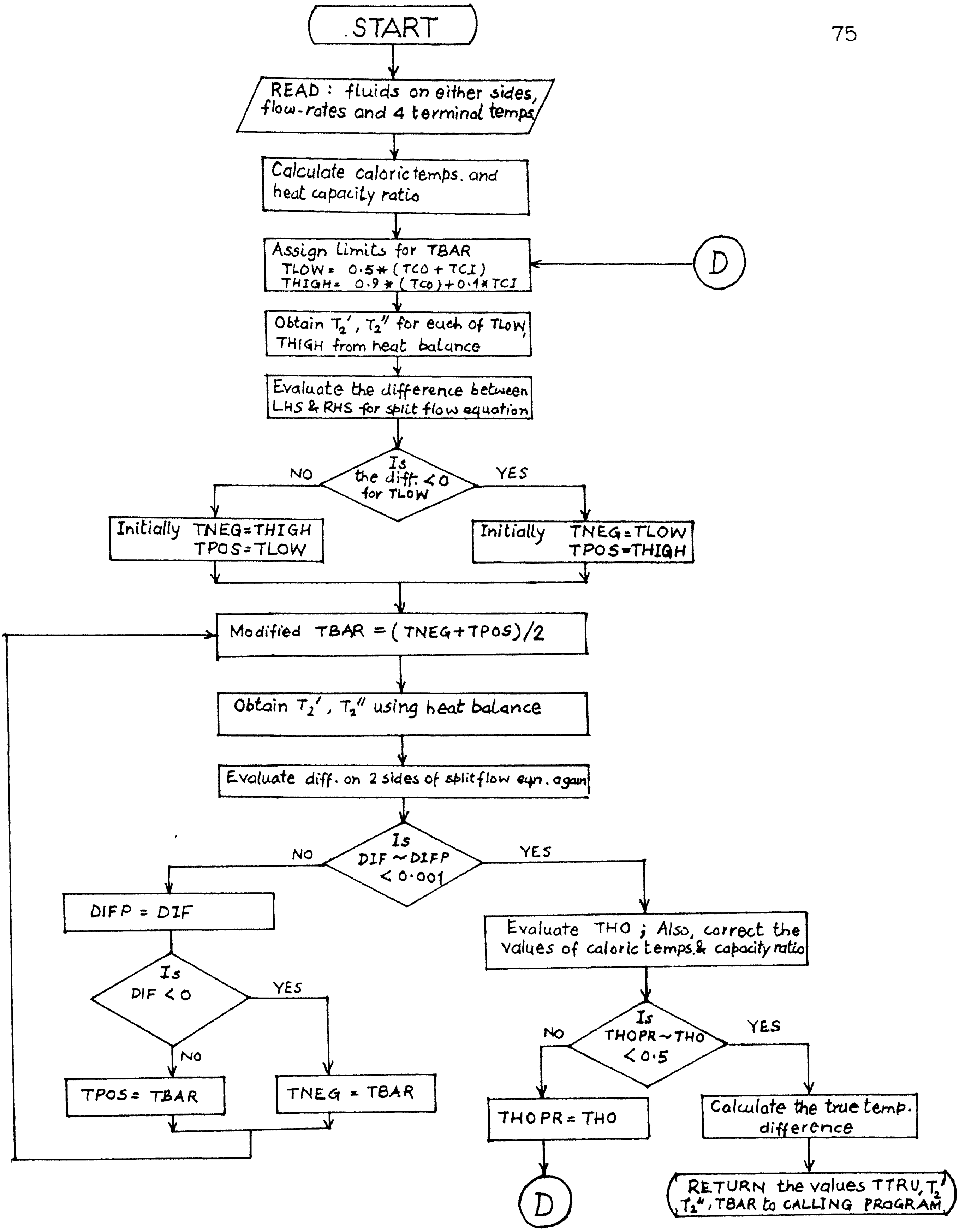
A - 6: SUBROUTINE HTCOEF



A-7: SUBROUTINE PRDROP



A-8 : SUBROUTINE TRUTEM



A-9: SUBROUTINE SPFRTH

DESIGN AND PERFORMANCE EVALUATION

The terminal temperatres for counterflow case
HOT FLUID :
Inlet t= 50.00 Outlet t= 35.00

COLD FLUID :
Inlet t= 5.00 Outlet t= 21.93
The LMTD (counterflow)is 29.024 deg. C.

Max. No. of tubes = 565 ; actual tube count= 260
O.D. of tubes= 2.54 ; I.D.= 2.06 ; Length= 4.00
Internal dia. of shell= 63.500
Heat duty= 3767192.200 ; overall ht.tr. coeff.= 1564.030

No. of baffles(half shell)= 12 ; total no. of baffles= 25
Baffle spacing= 15.38

CONFIGN.	SIDE	EQ.DIA	REYNOLDS NO.	VELOCITY	PR.DROP	HT.TR.COEFF.
1	1	2.057	10604.4	0.614	5908.39	1974.93
1	2	2.513	121744.1	3.099	16260.50	9163.60

The coeffs for case 1 are 1564.03 1624.76

The terminal temperatures for splitflow case
HOT FLUID :
Inlet t= 50.00 Mean outlet t= 36.17
T2" = 34.47 T2"" = 37.86

COLD FLUID :
Inlet t= 5.00 Outlet t= 20.61
Mid-pt. t= 13.76

The true temp.-diff(splitflow) = 29.16

CONFIGN.	SIDE	EQ.DIA	REYNOLDS NO.	VELOCITY	PR.DROP	HT.TR.COEFF.
2	1	2.057	10417.3	0.614	5913.75	1958.86
2	2	2.513	61506.5	1.550	2316.13	5278.76

The coeffs for case 2 are 1411.62 1493.05

The dirt-factor for the exchangers is 0.000023898

COMPARISON AND MODIFIED DESIGN

Ratio of ht. duty = .926 ; Percent reduction 7.390
Actual pressure as fraction = .1424 ; deviation = 0.0174
Percent deviation = 13.95

1) If the length of exchanger is increased :
No. baf= 25 spc=16.6122 delp= 2019.32 %de= -0.65 ratio= 0.1242

2) If shell dia. and no. of tubes is increased :
No.tub= 300 shd=68.5800 delp= 2180.11 %de= 7.26 ratio=0.1341

TEMPERATURE VARIATION ALONG LENGTH OF EXCHANGER

Pt.no.	Locn.	TH(x)	Mf.(hot)	TC(x)	Mf.(cold)
		CONFIGURATION NO. 1			
1	0.000	50.00	60.00	21.93	53.00
2	0.500	48.18	60.00	19.88	53.00
3	1.000	46.34	60.00	17.80	53.00
4	1.500	44.49	60.00	15.71	53.00
5	2.000	42.62	60.00	13.61	53.00
6	2.500	40.74	60.00	11.48	53.00
7	3.000	38.84	60.00	9.34	53.00
8	3.500	36.93	60.00	7.18	53.00
9	4.000	35.00	60.00	5.00	53.00
		CONFIGURATION NO. 2			
1	0.000	34.47	60.00	5.00	53.00
2	0.500	38.07	60.00	7.03	53.00
3	1.000	41.87	60.00	9.17	53.00
4	1.500	45.86	60.00	11.43	53.00
5	2.000	50.07	60.00	13.80	53.00
6	2.500	46.06	60.00	15.98	53.00
7	3.000	42.79	60.00	17.83	53.00
8	3.500	40.08	60.00	19.36	53.00
9	4.000	37.83	60.00	20.63	53.00

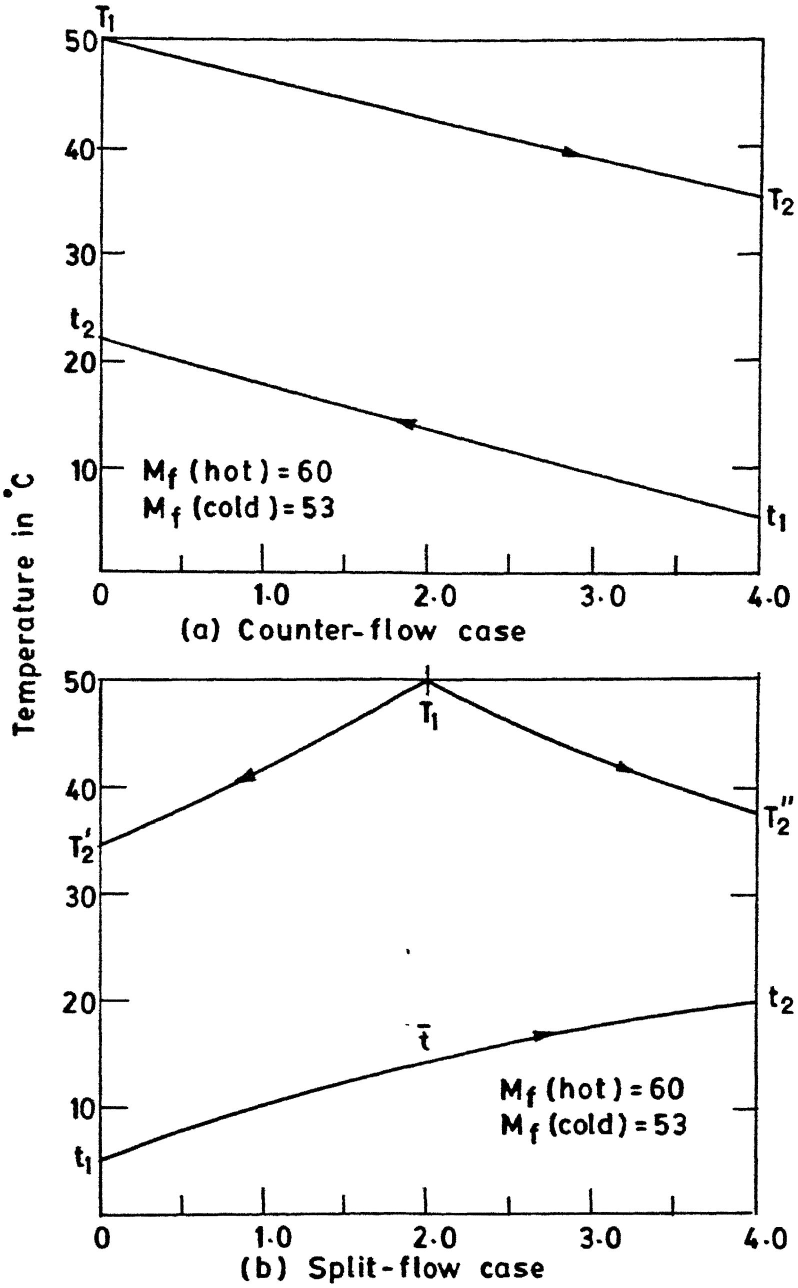


FIG. A-10 TEMPERATURE VARIATION ALONG LENGTH FOR SAMPLE SOLUTION